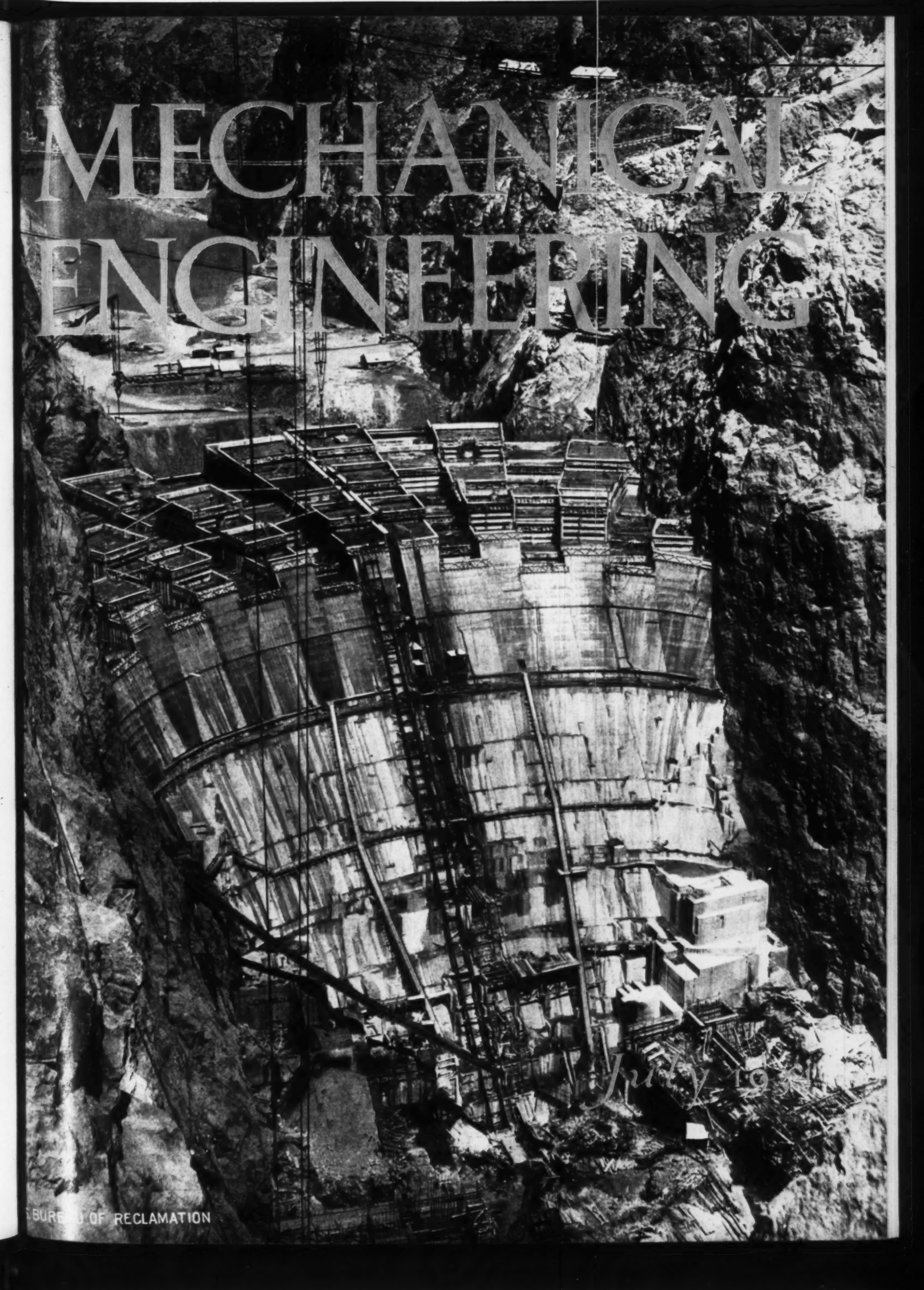
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July 1951
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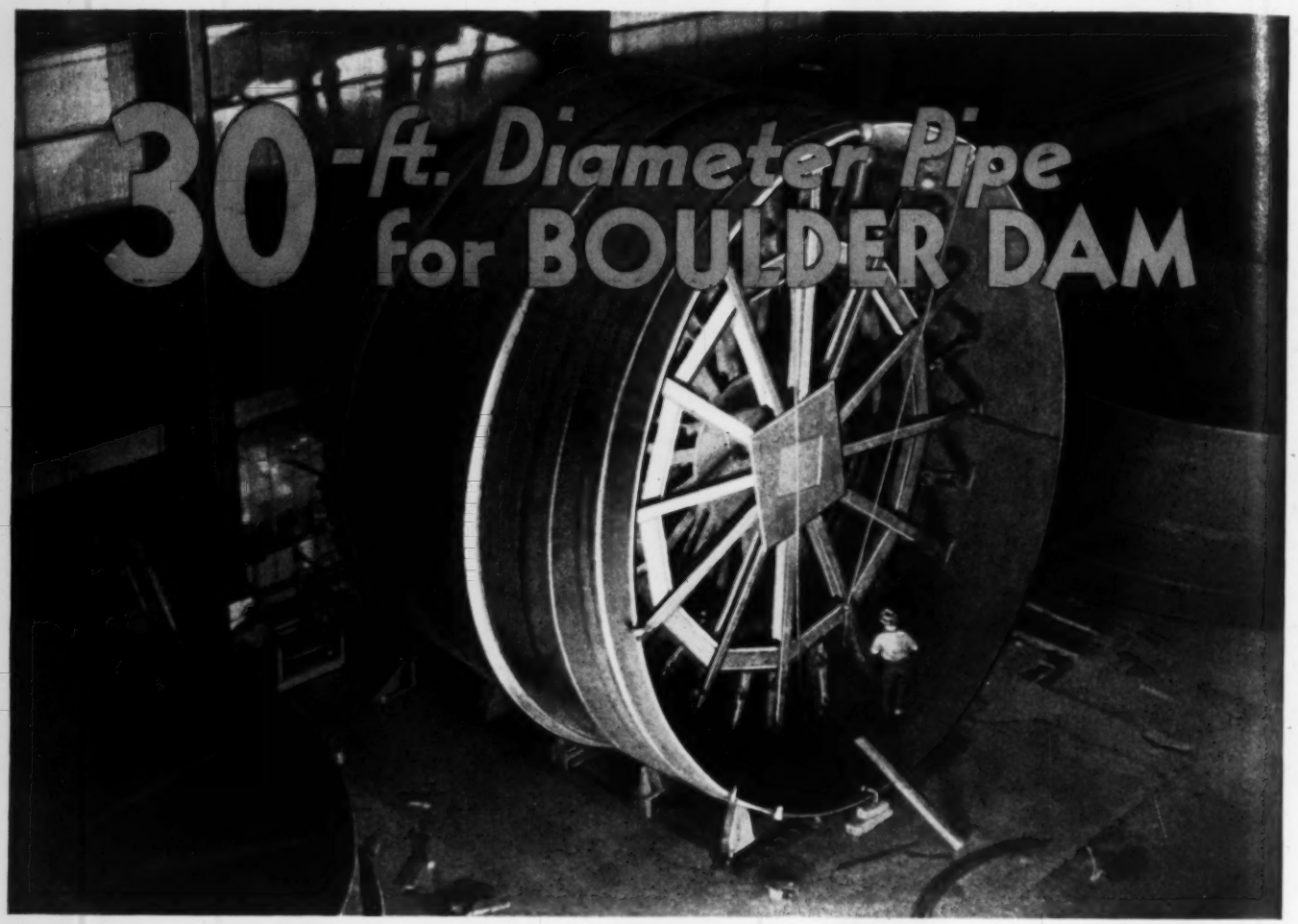

Under the second largest contract ever awarded by the United States Department of the Interior, The Babcock & Wilcox Company is forming approximately 45,000 tons of steel into 14,500 ft. of pipe for Boulder Dam.

As existing railroads cannot transport pipe sections as large as the 30-ft. diameter section shown above, a complete fabricating plant has been built by The Babcock & Wilcox Company near the Dam. Here steel plates are planed, formed, and welded into erection-sections of pipe . . . the welds X-rayed for soundness and each section stress-relieved.

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The Babcock & Wilcox Company . . . 85 Liberty St. . . . New York . . . N. Y.

BABCOCK & WILCOX

MECHANICAL ENGINEERING

Published by The American Society of Mechanical Engineers

VOLUME 56 — NUMBER 7

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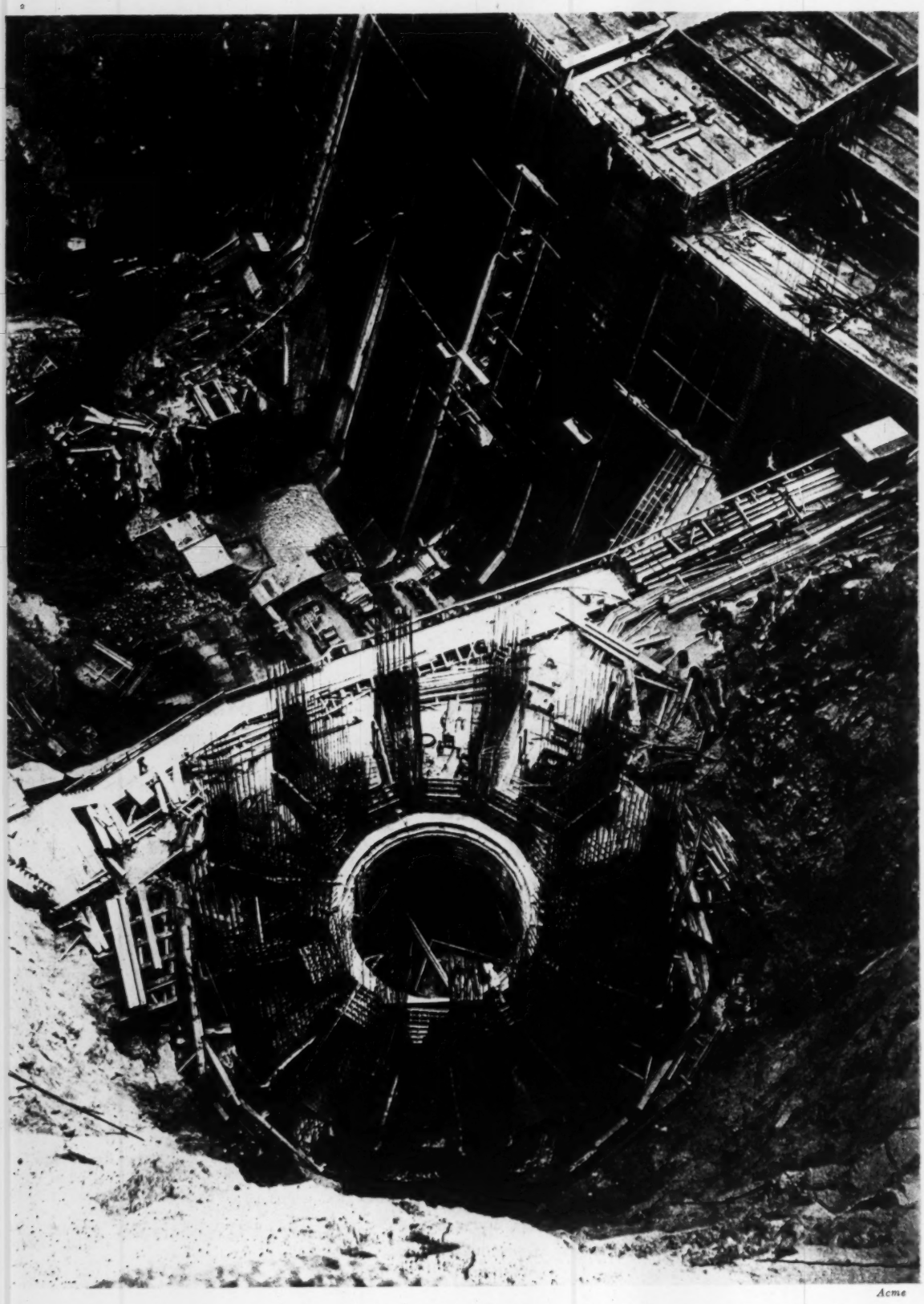


Acme

Starting the Construction of One of the Four Intake Towers at Boulder Dam

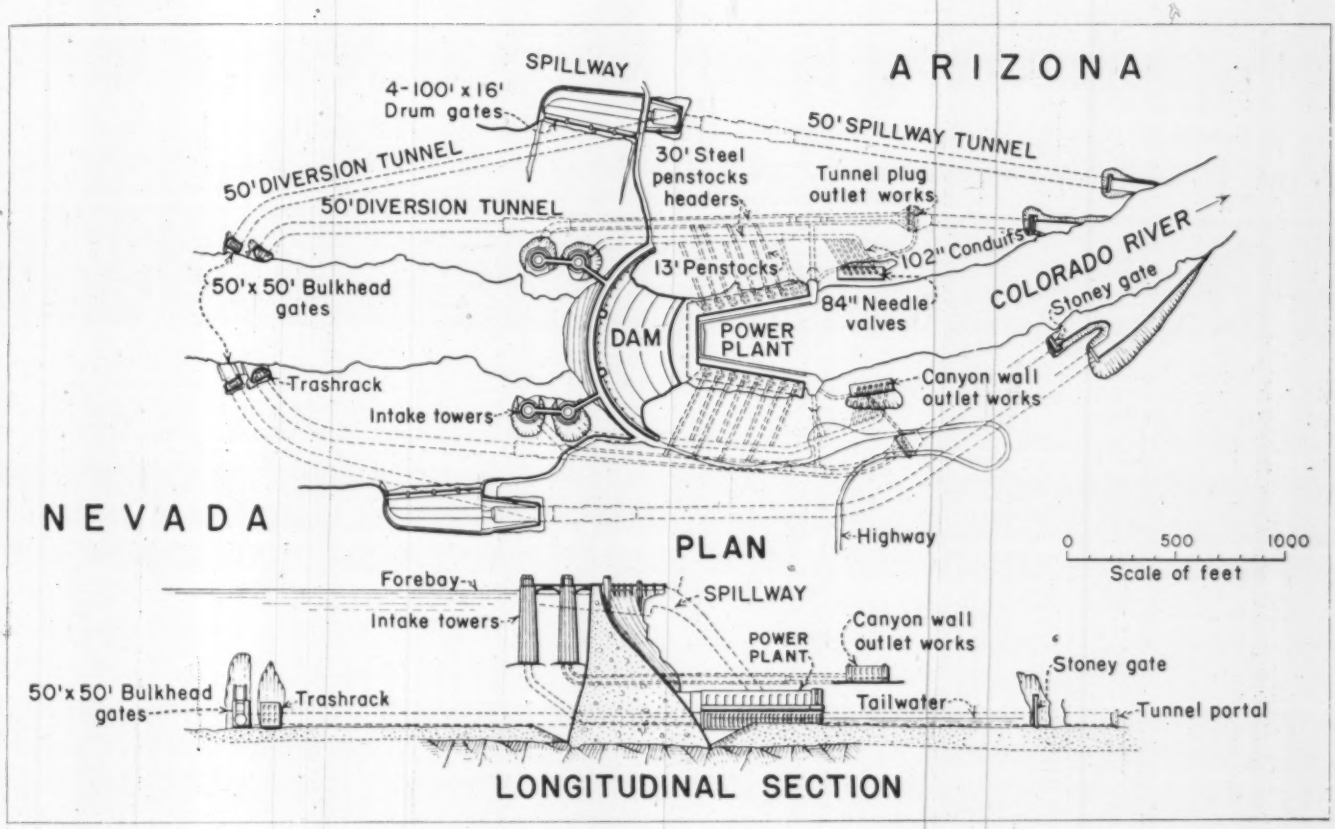


FIG. 1 GENERAL LAYOUT, BOULDER DAM AND ASSOCIATED WORKS

HYDRAULIC VALVES *and* GATES *for* BOULDER DAM

Part I—The Design and Fabrication of the Cylinder Gates and Operating Mechanism

By P. A. KINZIE[1]

THE INTAKE TOWERS

EACH OF THE four intake towers in the reservoir adjacent to the upstream face of Boulder dam, as shown in Fig. 1, is provided with two cylinder gates, 32 ft in diameter, which are disposed vertically one above the other in such a manner that the lower gate is placed in the base of the tower at elevation 895 and the upper gate 150 ft higher. See Fig. 1*a*.

These gates will be used as shut off gates for the plate-steel outlet pipes (penstock headers), 30 ft in diameter, that extend from the base of each tower to the turbines in the power plant and to the batteries of needle valves in the canyon walls below the dam and also to the batteries of needle valves in the tunnel-plug outlet works. Through these gates will pass all irrigation water released through the needle valves, and all water entering the turbines in the power plant.

By closing the upper and lower gates in any one intake tower the 30-ft outlet pipe connected therewith can then be drained for internal inspection and maintenance work. This procedure will likewise unwater and permit access to the interior of four 13-ft penstocks and to the upstream side of four of the butterfly valves in the power plant, and also to six of the Paradox emergency gates protecting the needle valves in the outlet works.

Inasmuch as the closure of the gates in a single tower will entail shutting down 25 per cent of the power plant by taking four 115,000-hp turbines with their generators off the line with a consequent loss of power revenue of approximately $4500 every 24 hours, it is evident that it will come near to requiring an act of Congress to sanction such a procedure once this equipment has been placed in regular service.

With the water surface in the reservoir at its maximum eleva-

[1] Engineer, U. S. Bureau of Reclamation, Denver, Colo.

For presentation on "Reclamation Bureau Night," at the Semi-Annual Meeting, Denver, Colo., June 25 to 28, 1934, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Published by permission of the United States Bureau of Reclamation.

tion of 1229, the static head on the center of the upper gates will be 179 ft, and on the center of the lower gates it will be 329 ft. Each gate will be raised to open and lowered to close by means of three stems equally spaced around the gate's circumference. These stems extend upward to powerful electric-motor-driven hoists in the operating rooms at the top of the intake towers.

The four intake towers are placed in pairs in semi-circular recesses blasted out of the rock walls of the canyon, two on the Arizona side of the river and two on the Nevada side. The rock benches upon which the bases of these towers rest were formed by blasting operations, which required the removal of 360,000 cu yd of solid rock, more material than is contained in many well-known dams. The benches are all on a substantially common level, 388 ft above the lowest portion of the dam's foundations in the river gorge below.

The twin sets of towers rising from these benches are identical in size and shape, but the relative positions of the gate hoisting equipment in the tops of the towers are altered slightly to preserve symmetry with respect to the bridges which connect the operating rooms of each pair of towers and then extend downstream to the top of the dam.

By referring to Fig. 1, a clear idea of the relative positions of these intake towers with respect to the dam and its principal associated control works will be readily acquired. Fig. 1 likewise clearly shows the inter-relationship of these towers with the four 30-ft steel penstock headers which carry the water from the reservoir to the turbines to generate power, and to the needle valves to be released into the river for use on the thirsty lands far down the river valley. The trashracks on these intake towers remove from the water of the reservoir, as it enters the towers, its burden of logs, sticks, and debris, so that any danger of clogging the wicket gates or the needle valves, or injuring the turbine runners or other machinery affected, is averted.

These reinforced concrete intake towers are 348 ft high, as measured from the rock benches on which they rest to the hoist-room floors, level with the roadway across the top of the dam, at elevation 1232. The hoist houses rise 48 ft above their floors, and the tips of the cast-aluminum lanterns on their domes are 55 ft above the operating-room floor, making the total height of the towers 403 ft. Within the center of each tower is a vertical cylindrical shaft or barrel, 29 ft 8 in. inside diameter, with enlargements at the lower and upper gate levels to enclose and house the two cylinder gates installed in each tower.

Projecting radially from the shaft or barrel walls of each tower are 12 concrete fins or buttresses. (See sectional plan and section *C–C* of Fig. 1*a*.) These fins add the necessary stability to the towers and form the inwardly extending water channels between the trashracks, adjacent to their outer extremities, and the entrance liners embedded in the shaft walls leading to the cylinder gates within the towers. They are also provided with grooves which act as guides for the emergency bulkhead gates when these are being lowered to close off the inlet openings of the entrance liners leading to the cylinder gates. The trashracks are installed between these fins so that the exteriors of these towers virtually are giant screens.

The outside diameter of the finned towers is 82 ft at the base and about 64 ft at the operating floor level, giving a batter of 1 in 39 to the structure as a whole, including the trashracks. The outside face of the shaft wall is also battered to provide greater thickness at the bottom. The thickness of the shaft walls was designed primarily to withstand the external water pressure, but the entire towers were designed to withstand earthquake stresses as well.

To reinforce the towers adequately against seismic disturbances involved the use of an immense amount of reinforcement steel, the major portion of which was necessarily of a hard grade. Most of the vertical steel was purchased in 57-ft lengths and lapped by spacing a minimum of 6 in. apart to develop the full bond strength of the concrete and steel. As

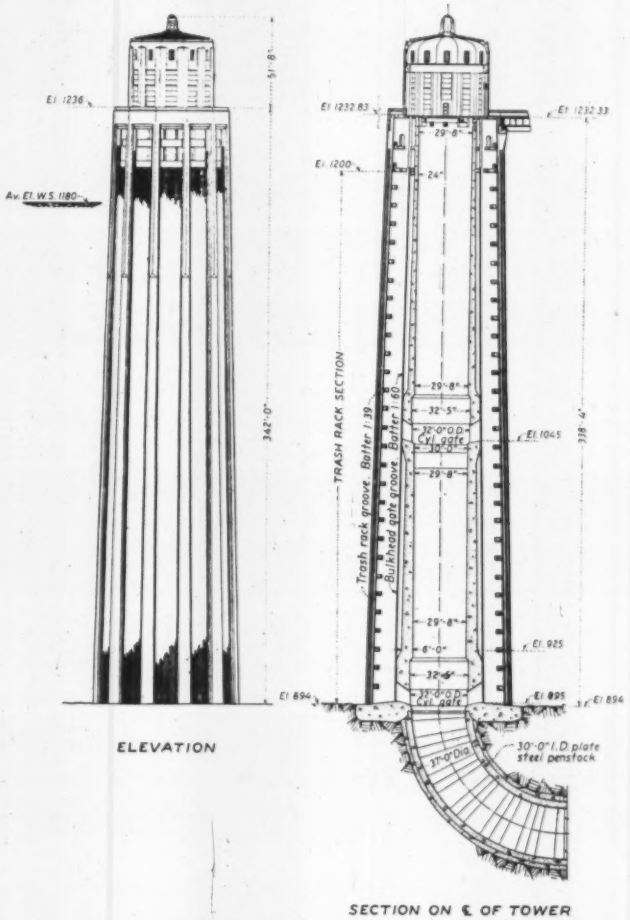
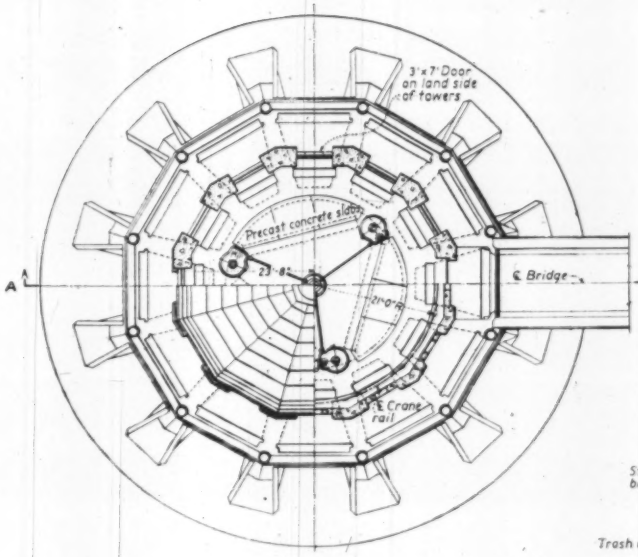


FIG. 1*a* INTAKE TOWER, SECTIONS

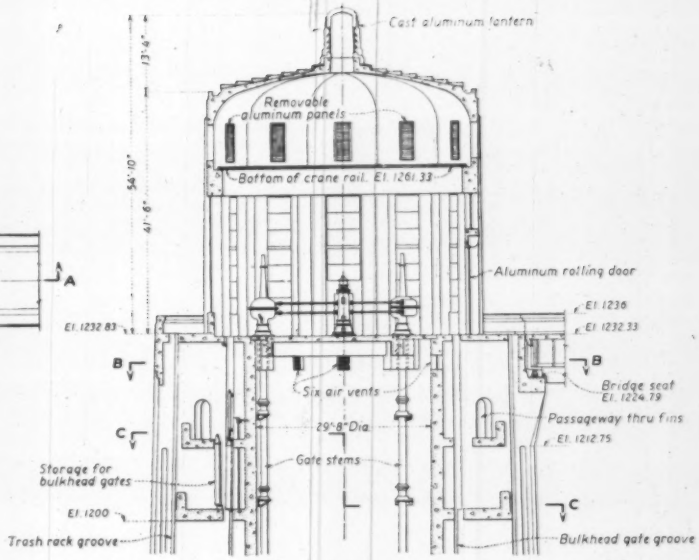
finally designed and constructed, an average of 158 lb of reinforcing steel per cubic yard of concrete was used.

TRASHRACKS

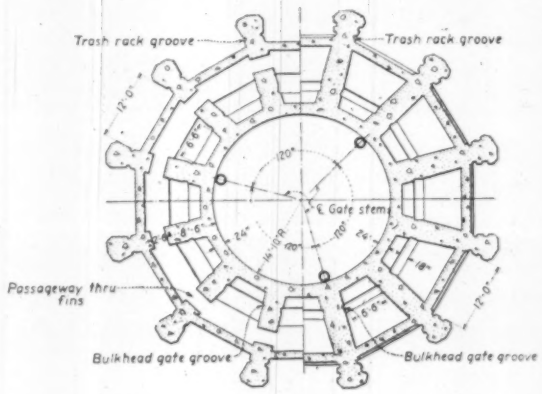
At the elevations corresponding to the levels of the upper and lower gates, the intake tower walls are pierced by 12 equally spaced and radially converging waterways as shown in Fig. 2. As the inflowing water approaches these waterways through the tower walls to the interior of the towers, it passes through the vertical bars of the trashracks, which extend from the base of each tower to elevation 1200. These trashracks enclose the spaces between the radial buttress walls or fins of the towers and are inserted in grooves provided for that purpose near the outer extremities of the opposing buttress wall faces. By this arrangement every tower is provided with 12 sets of trashracks 12 ft $7^{3}/_{4}$ in. wide by 306 ft $6^{1}/_{2}$ in. high equally spaced around their circumferences. These racks consist of specially rolled 5 by $^{3}/_{4}$ in. steel bars with half-round edges to improve flow conditions, spaced horizontally on $4^{7}/_{8}$-in. centers, jig assembled so that they will stack one above another with even bearing, and then welded to their horizontal spacers. Two types of trashrack units are used. The bottom units are 10 ft



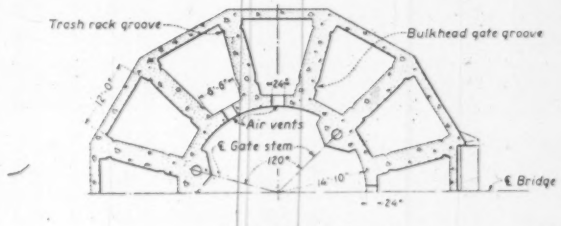
SECTIONAL PLAN



SECTION A-A



SECTION C-C



SECTION B-B

AND ELEVATIONS

$2\frac{1}{2}$ in. high and weigh 4480 lb, and the regular units are 10 ft 7 in. high and weigh 4620 lb each. Each of the 12 sets of trashracks required for one tower consists of one bottom section and 28 regular sections stacked end on end vertically in the grooves in the buttress walls of the towers. In all, 1398 units, with a total weight of 6,426,000 lb, are required for the four towers. These trashracks were fabricated by the Ingalls Iron Works, of Birmingham, Ala., under government plans and specifications No. 547.

The gross area of these trashracks for one tower is 46,452 sq ft, and for all four towers it is 185,808 sq ft. The total net area between the bars is 141,494 sq ft. An average city block is 300 ft long by 325 ft deep (allowing 25 ft for the alley) and it contains 90,000 sq ft of ground, exclusive of the alley. Using this as a measure of comparison, the total area of the trashracks on all four intake towers would cover two such city blocks and about one-third of the intervening street between them; or expressing this in another way, they would cover four and one-quarter acres of ground.

With the power plant operating at full capacity and with the reservoir water surface at elevation 1221.4, the velocity through the racks will be 0.216 fps, and with the power plant and the needle valves in the outlet works all operating at maximum capacity the velocity through the racks will then be 0.848 fps.

With the reservoir water surface at elevation 1150 and the power plant operating at full capacity, the velocity through the racks will be 0.327 fps, and with the reservoir water surface at elevation 1070 this velocity will then become 0.522 fps. These velocities are all based upon the total net effective areas of the racks for the respective reservoir water surfaces. In some instances the actual velocities through the racks will be somewhat higher than the values given.

The specially rolled steel bars used in the trashracks have the following physical properties: Ultimate tensile strength, 75,000 lb per sq in.; yield point, 40,000 lb per sq in.; and elongation in 2 in., not less than 16 per cent.

The trashrack beam and seat plates embedded in the concrete of the towers on which the trashracks rest were purchased on government plans and specifications No. 593-D and require a further addition of 598,000 lb of castings and other metal work.

INLET PASSAGES

The inlet passages thus guarded by the trashracks are formed by semi-steel entrance liners embedded in the concrete walls of the intake towers. The inwardly facing end flanges of these liner castings are bolted to the outwardly facing and complementary flanges of the welded plate-steel nose liners within which each cylinder gate is guided and held centrally with respect to the vertical cylindrical shaft inside each tower.

The nose liners rest upon and are bolted to the upper flange faces

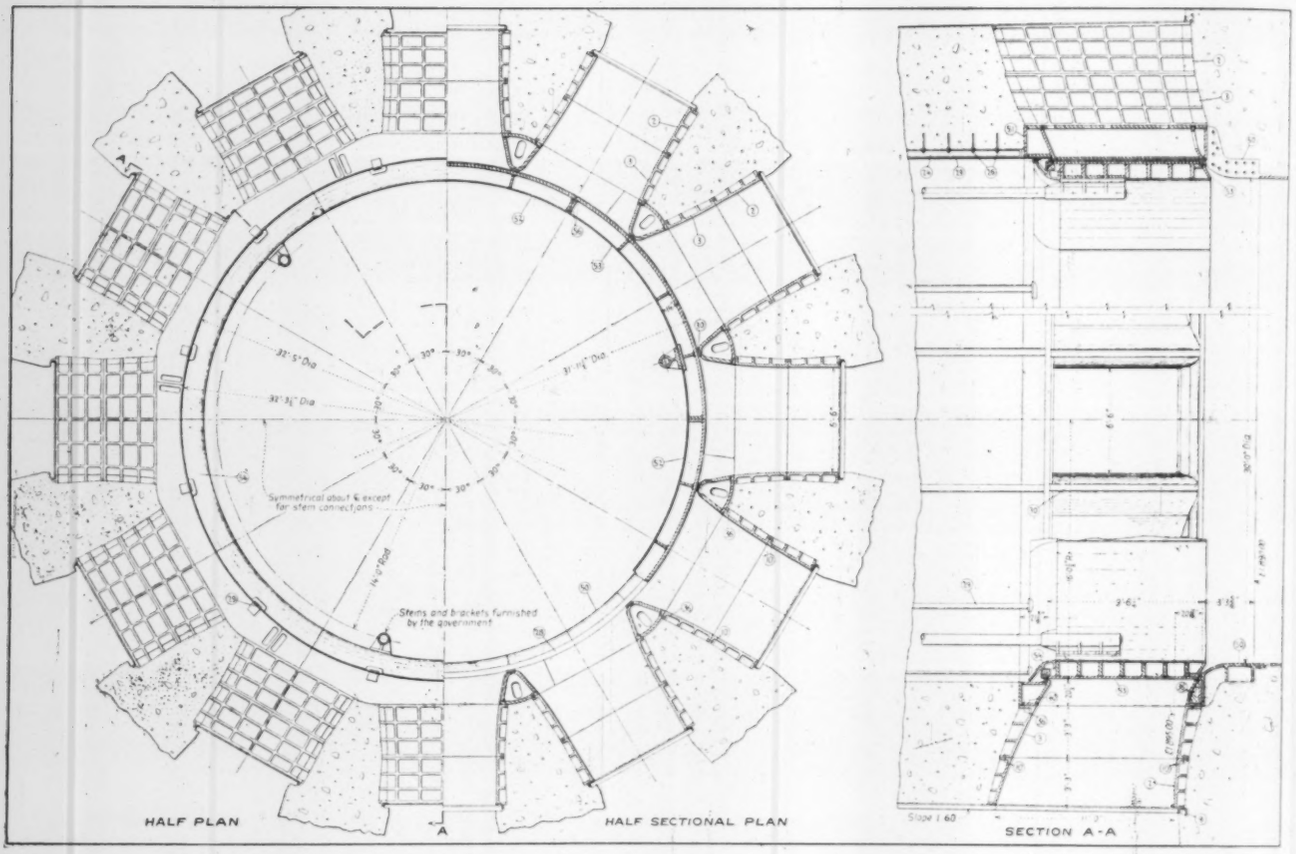

HALF PLAN
HALF SECTIONAL PLAN
SECTION A-A
Symmetrical about ℄ except for stem connections
Stems and brackets furnished by the government

FIG. 2 GENERAL ASSEMBLY—LOWER GATE

of the throat liners which are also solidly embedded in the tower concrete as may be seen in Section *A–A* of Fig. 2. The curving surfaces of these water passages are so arranged that the water flows smoothly into the interior tower shafts and downward into the 30-ft pipes below. These shapes were mathematically established and then checked by model tests in the hydraulic laboratories of the Colorado Agricultural College at Ft. Collins. When working to the maximum capacity of the turbines and needle valves, the four intake towers will be passing a total of 120,000 cfs, and the potential energy in this water will then be approximately 8,000,000 hp.

The inwardly converging nose liners terminate in rounded wedge-shaped nosings 9 ft 6 in. high as shown in the half-sectional plan of Fig. 2, which enclose a circle 32 ft $^1/_2$ in. in diameter. Each of these vertical nose faces is provided with inserted bronze facing strips 5 in. wide which form the vertical guides within which each cylinder gate travels upward or downward as the case may be to uncover or cover the water passages between the noses and so control the flow of water into the towers.

THE CYLINDER GATES

Each cylinder gate is a cylindrical plate-steel shell. The outside diameter of the upper gates is 31 ft 11 in., their height is 9 ft $^1/_2$ in., and the plates are $1^1/_2$ in. thick; for the lower gates the outside diameter is 31 ft $11^1/_2$ in., the height 9 ft $^1/_2$ in., and the plate 2 in. thick.

These vertical cylindrical shells are supported from within against the external pressure of the water by heavy ring ribs. In the upper gates these rings are $11^1/_2$ in. deep by $1^1/_2$ in. thick spaced 16 in. apart vertically; in the lower gates they are $11^1/_4$ in. deep by $2^3/_4$ in. thick on 16 in. vertical centers, as shown in Fig. 3. Vertical stiffener ribs $11^1/_4$ in. deep by $1^1/_2$ in. thick spaced 15 deg apart circumferentially are provided between the ring ribs of both upper and lower gates.

The bottom rings of both upper and lower gates are formed into inwardly and downwardly converging conical surfaces facing at an angle of $27^1/_2$ deg from the horizontal, so that when the gates are raised to their wide-open positions their underring surfaces will then be in continuity with the downwardly curving ceilings of the adjacent water passageways through the nose liners.

The upper gates weigh 150,000 lb each and the lower gates 240,000 lb. The material required for the gates, nose, throat, and entrance liners, is listed in Table 1.

TABLE 1

Copper-bearing steel plate, lb	3,440,000
Structural-steel shapes, lb	14,500
Welding rod, lb	20,000
Steel castings, lb	331,320
Semi-steel castings, lb	3,018,680
Bronze castings and strips, lb	87,500
Monel-metal seats, lb	111,240
Monel-metal bolts and screws, lb	15,000
Stainless-steel bolts, lb	34,000
Steel bolts, studs, and nuts, lb	30,000
Total, lb	7,102,240

Since water entering the intake towers beneath the upper gates must flow downward and through the lower gates, a $^3/_4$-in. liner plate, forming a cylinder 29 ft $9^1/_2$ in. inside diameter, is attached to and covers the lower gate's interior ribs so that the downwardly flowing water is afforded a smooth and uninterrupted passage. For the same reason a set of segmental

steel castings is bolted to the upper faces of the lower gates to form bell-mouth deflectors at the upper side of each gate. These deflectors also protect the upper seats of the bottom gates from the direct impact of the descending water.

Each of the upper and lower gates is divided into six equal segmental sections with vertical, radial flanged faces machined to bear and drilled for $1^3/_4$-in. and 2-in. body bound bolts through $1^3/_4$-in. and 2-in. flanges in the upper and lower gates, respectively. The upper gate segments complete weigh 25,100 lb each, while the lower gate segments weigh 31,600 lb each.

The details of the lower cylinder gate segment are shown in Fig. 4, while the details of the lower nose liner segment are shown in Fig. 5.

Each gate is provided with a fixed seal on its bottom end, consisting of a monel-metal ring set into a groove machined in the gate, and an adjustable seal ring of monel metal set into a machined recess around the upper end. These male seals engage female seat rings of monel metal set into machined recesses in the nose liners as may be seen in Fig. 3.

An upwardly extending framework, consisting of 12 vertical semi-steel castings of typical H-column section with buttressed bases finished and bolted to mating finished pads on the tops of the nose liners and with their upper ends receiving bolted heavy gusset plates to which a horizontal spacing ring formed from 4-in. by 4-in. by $^3/_8$-in. angles is riveted, is embedded in the concrete walls of the intake towers above each gate so that the inwardly opposing faces of the vertical column members are exposed and flush with the surface of the concrete. This framework is shown in place in Fig. 6, on the completed shop assembly of the first gate. Into the exposed faces of these embedded columns rolled-bronze guide bars 5 in. broad by $^7/_8$ in. thick are inlayed into finished recesses or seats and held in place by flathead machine screws $^7/_8$ in. in diameter. Twelve semi-steel guide-shoe brackets are bolted at equal intervals around the top of each gate opposite these vertical bronze guide bars and carry bronze guide shoes which are made to form an easy sliding fit on the guide bars and thereby hold the gates central with the tower shafts and at the same time prevent any rotational movement of the gates.

Limitations of Weight. As originally designed, each gate was made up with 12 segments of steel castings bolted together. This involved much machine work in the flanged joints and the hazards of concealed defects were present. This type of construction was also very heavy and consequently the hoists had to be correspondingly large and of high capacity. Inasmuch as one of the fundamental requirements in design was to make the intake towers capable of withstanding earthquake shock, and because, on account of their exceptional height (396 ft), tremendous quantities of reinforcing steel were necessary in order to satisfy this requirement, it was essential to reduce the weight in the tops of the towers to the lowest practicable amount. Confronted with these conditions, it was evident that every legitimate means available should be employed to reduce the weight of the cylinder gates so that the hoisting loads and capacities and consequently the weights of the hoists themselves in the tops of the intake towers might be lessened and so permit the enormous cantilever reinforcing in the towers to be correspondingly reduced.

A very small part of this reinforcing steel may be seen in Fig. 7; also in Figs. 21, 22, and 23. As finally designed for the reduced weights of cylinder-gate hoists and stems, it was found that 14,700,000 lb of reinforcing steel was required for the four intake towers.

To reduce the weight of the gates it was evident that the so-called "ignorance factor" allowance would have to be eliminated, or, to express this in another way, the material employed in the design and construction of these gates would have to be of such nature as to remove all uncertainties as to concealed defects, non-uniformity, and maintenance of tolerances. It was also apparent that to carry this weight-saving campaign to its logical conclusion, the number of radial flanged joints should be reduced until the pieces were just within the limits of size which it would be practicable to ship.

Welding Substituted for Casting. Examination of these new conditions revealed the necessity of discarding steel castings and making up new designs, and a further survey showed that high-quality steel plate would amply satisfy most of the remaining conditions, including the increased circumferential lengths involved in reducing the number of segments from twelve to six, while still maintaining the close tolerances required.

The massive size of these gates necessitated the use of such heavy plate sections that joining by conventional riveting or pinning was not practicable. It was then evident that welding by the most up-to-date methods was the one workable solution to this problem. Welding offered the advantages of uniformity of material, ease of inspection, practical elimination of concealed defects such as hidden blow-holes and shrinkage cracks, ability to shape the parts by pressing and rolling to meet the close tolerances required without excessive machining, and the elimination of time and cost required for the preparation of patterns.

The decision was made, therefore, to take advantage of the rapid advance in the art of welding as developed within the last few years, and the designs described and illustrated in this paper are the results of that decision.

DESIGN OF THE STEEL-PLATE WORK

With the ruling conditions established as described and the desirable features definitely in view, new designs were started based upon those conditions and the following data:

The steel plates were to have an ultimate strength and elastic limit of 65,000 and 35,000 lb per sq in., respectively; the allowable stresses in tension and shear, 16,000 and 12,000 lb per sq in., respectively; and the allowable stress in compression $16{,}000/(1 + l^2/13{,}500r^2)$ lb per sq in.

The maximum working pressure to which the gates might eventually be subjected was increased by 25 per cent for impact.

The ring section of both upper and lower gates was so designed as to provide a factor of safety of 4, plus or minus; i.e., 4 times maximum working pressure equals collapsing pressure.

In the gate-ring sections selected, the following stress analyses were made: Direct ring compression, plate bending between stiffener rings, distortion bending of segment rings, and critical combinations of these stresses. Each gate ring is made up of six segments bolted together. Combined stresses in these bolted joints were investigated.

The formulas[2] used in computation follow:

For plate bending

$$M = Fs = 1/12\ wl^2;\quad \Delta = wl^4/192EI$$

For direct ring compression

$$f = wbR/A$$

For distortion bending

$$u = \frac{R^3P}{EI}\left(\frac{1}{\pi} - \frac{\pi}{8}\right);\quad \text{when } \phi = \frac{\pi}{2}$$

$$u = \frac{R^3P}{EI}\left(\frac{1}{\pi} - \frac{1}{4}\right);\quad \text{when } \phi = 0$$

$$M = -0.318\ PR$$

[2] See Timoshenko, "Strength of Materials," vol. 2, pp. 437, 460, 601.

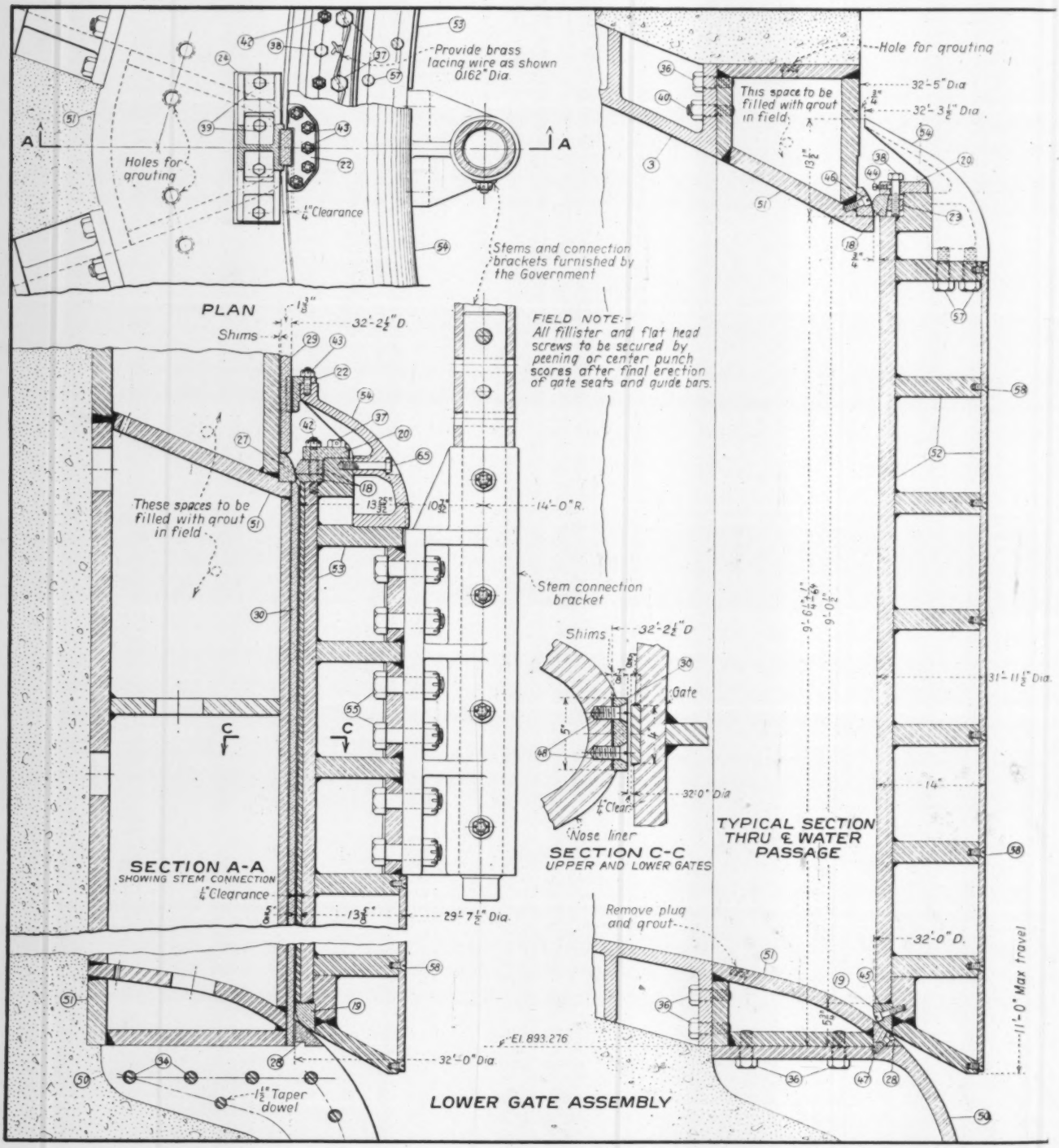

PLAN
SECTION A-A
SHOWING STEM CONNECTION
SECTION C-C
UPPER AND LOWER GATES
TYPICAL SECTION
THRU ℄ WATER
PASSAGE
LOWER GATE ASSEMBLY
Provide brass lacing wire as shown 0.162" Dia.
Holes for grouting
Stems and connection brackets furnished by the Government
FIELD NOTE:-
All fillister and flat head screws to be secured by peening or center punch scores after final erection of gate seats and guide bars.
Hole for grouting
This space to be filled with grout in field
These spaces to be filled with grout in field
Stem connection bracket
Shims
Gate
Nose liner
Remove plug and grout
1½" Taper dowel
El. 893.276
11'-0" Max travel

FIG. 3 DETAIL ASSEMBLIES

For the combined stresses: Uni-directional stresses of like sign are added directly; $z \times$ a cross stress of a given sign is added to a direct stress of opposite sign.

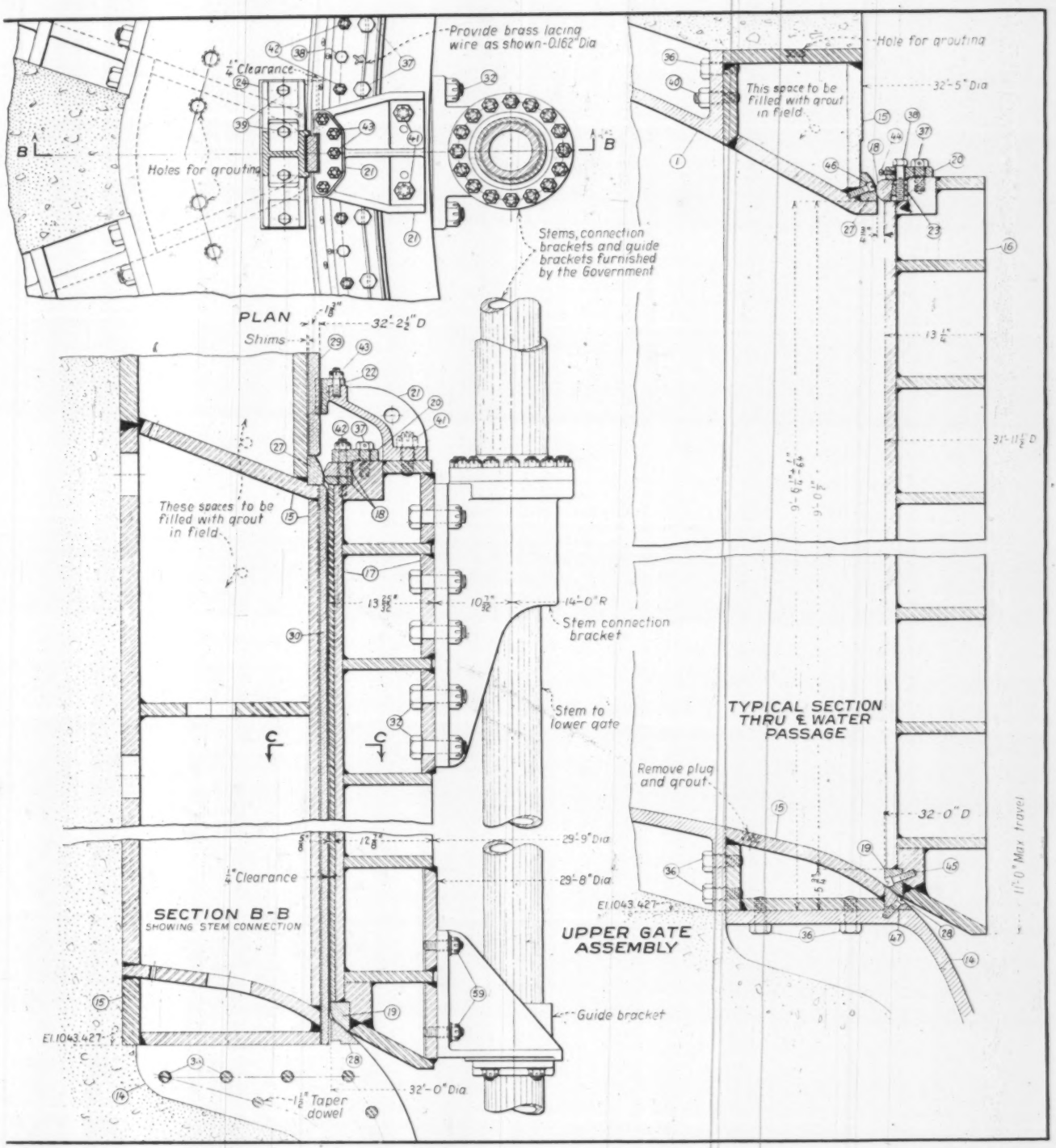
For the collapsing pressure

$$W = 3EI/bR^3$$

In the formulas quoted, f is the unit stress, lb per sq in., W the water pressure, lb per sq in., l the length of beam, in., b the breadth of beam, in., u the distortion or displacement, in., z Poisson's ratio (0.3 was used), P the applied pressure, lb per sq in., Δ the maximum beam deflection, in., S the section modulus (in.^{3}), I the moment of inertia (in.^{4}), E the 30,000,000, M the bending moment, in-lb, R the outside radius of ring, in., and A the area of section, sq in.

DESIGN PROCEDURE

Several types of ring sections were examined and finally sections with reinforcing rings substantially as shown in Fig. 4



—UPPER AND LOWER GATES

were selected as being the most desirable since they required the least amount of welding and the plate spans between the ribs were not excessive. It was decided to hold the plate span between the ribs to within 16 times the plate thickness and $1^1/_2$ times the rib depth, with rib thickness not less than one-eighth of the net rib depth, not including the thickness of the plate. Different combinations of rib and plate were analyzed and finally the one shown in Fig. 4 was chosen for the bottom gate.

The outside diameter of the ring was fixed by hydraulic and other conditions, while the minimum inside diameter of the bottom gate obviously should not be made less than 30 ft.

When examining the ring sections chosen it was apparent that if the rib sections were cut free from the cylindrical plate sections, the radial shortening of the former would be less than that of the latter. In the case of the lower gates the following data apply:

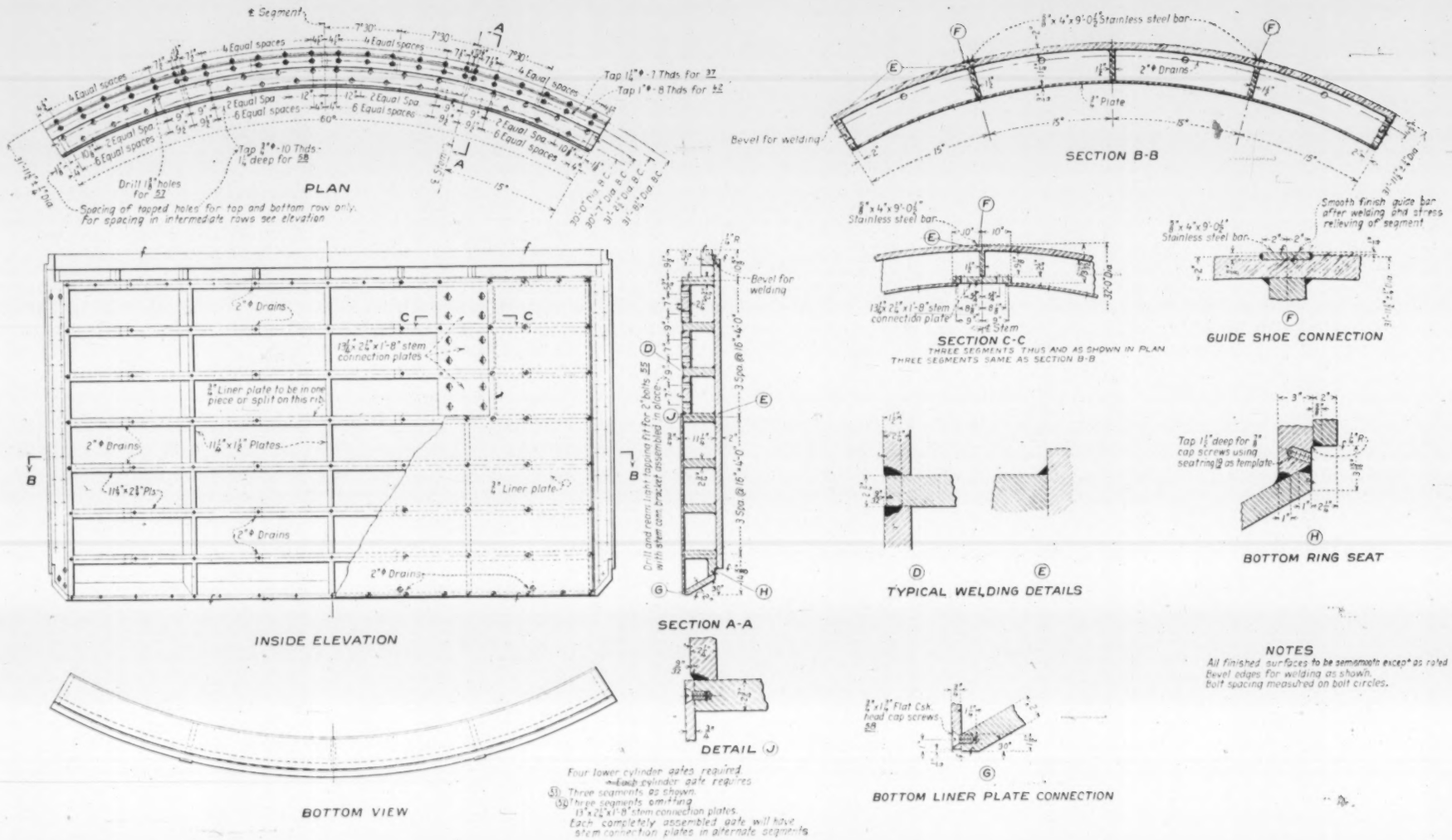

℄ Segment
PLAN
Tap 1¼"ø-7 Thds for 37
Tap 1"ø-8 Thds for 52
Tap ⅞"ø-10 Thds 1¼" deep for 58
Drill 1⅛" holes for 37
Spacing of tapped holes for top and bottom row only. For spacing in intermediate rows see elevation
INSIDE ELEVATION
2"ø Drains
Liner plate to be in one piece or split on this rib
1¼"x1⅛" Plates
⅞" Liner plate
BOTTOM VIEW
SECTION A-A
Bevel for welding
Drill and ream Light tapping fit for 2" bolts 55 with stem conn. bracket assembled in place
DETAIL (J)
Four lower cylinder gates required.
Each cylinder gate requires
(51) Three segments as shown.
(50) Three segments omitting 13"x2½"x1'-8" stem connection plates.
Each completely assembled gate will have stem connection plates in alternate segments
SECTION B-B
⅜"x4"x9'-0½" Stainless steel bar
2"ø Drains
1" Plate
Bevel for welding
SECTION C-C
THREE SEGMENTS THUS AND AS SHOWN IN PLAN
THREE SEGMENTS SAME AS SECTION B-B
13⅛"x2½"x1'-8" stem connection plate
Smooth finish guide bar after welding and stress relieving of segment
GUIDE SHOE CONNECTION
TYPICAL WELDING DETAILS
Tap 1½" deep for ⅞" cap screws using seating as template
BOTTOM RING SEAT
⅞"x1¼" Flat Csk. Head cap screws
BOTTOM LINER PLATE CONNECTION
NOTES
All finished surfaces to be semismooth except as noted
Bevel edges for welding as shown.
Bolt spacing measured on bolt circles.

FIG. 4 LOWER-CYLINDER-GATE SEGMENT

High-water elevation, ft	1232
Bottom of gate, ft	892
Head, ft	340
Equivalent pressure, lb per sq in.	147
Impact, 25 per cent, lb per sq in.	37
Total, lb per sq in.	184
Design pressure used, lb per sq in.	200

Then for the ribs alone, the radial shortening was 0.0185 in., and for the plate alone, the radial shortening was 0.123 in.

Also if ribs are considered as supports carrying the load from the plate, the deflection Δ at the center of each span between ribs is 0.0031 in. Since $\Delta = 0.0031/0.123 = 0.03$ of radial shortening, this factor was too small to affect appreciably the results and was neglected. The adjustment of load between the ribs and the plate was then investigated and also found to be negligible.

The ring compression was computed as follows:

$$\frac{675 \times 192}{13.25} = \frac{102 \times 192}{2} = 9800 \text{ lb per sq in.}$$

The gates were allowed a maximum clearance of $^1/_4$ in. in diameter between the vertical stainless steel bars on the outside of the gate and the bronze guide bars on the tips of the nose-liner wedges. Assuming that a force was applied to distort the ring so as to cause it to bear on the opposite guides, then u in the distortion formula becomes $^1/_8$ in. Then using the distortion bending formulas P was found to be 400 lb per sq in., and the resulting bending stress f was found to be 4100 lb per sq in. at the inner edges of the ribs and 1890 lb per sq in. at the outer edge of the skin plate. The ring distortion assumed for bending was that it became elliptical with a major axis diameter of 32 ft $^1/_4$ in. and the minor axis diameter of 31 ft $11^3/_4$ in.

These stress values were for the typical ring sections, while at the top and bottom of the gate the conditions were found to be somewhat altered, but further investigations revealed no marked change in combined stresses. No allowance was made for the strengthening effect of the top and bottom seats, nor for the deflectors.

At the segmental joints heavy bearing plates accurately fitted together were provided and 2-in. bolts were used in sufficient numbers and so located as to provide a resisting moment equivalent to the maximum bending moment that could be developed in those localities.

Stresses were calculated on the gross dimensions of the plate and ribs without deductions for corrosion, as ample provision was being made for routine maintenance of these gates.

The maximum combined stress in the lower gates was found to be 13,800 lb per sq in. in the ribs, and the collapsing pressure was found to be 820 lb per sq in. Dividing 820 by 200 (design pressure, including 25 per cent allowance for impact) gives a factor of safety of 4.1.

As a matter of routine checking to insure correctness of the designs, the final designs were subjected to the following tests:

The formula for the critical external pressure on a circular body, as calculated for a thin ring, is given in "Strength of Materials," by Timoshenko, page 599, as follows:

$$Q_{cr} = 3EI/bR^3$$

where Q_{cr} = critical pressure, lb per sq in.
E = Young's modulus
I = moment of inertia of section
b = width of section, in.
R = mean radius of section, in.

In the calculation of the stresses in a given section, this external pressure produces a purely compressive stress which can be computed from the following formula:

$$f_c = QR/A$$

where f_c = compressive strength, lb per sq in.
Q = external pressure, lb per sq in.
R = radius of section, in.
A = area of section, sq in.

As it would not be practicable to manufacture cylinders 32 ft in diameter without some ellipticity unless they were completely turned and bored, an investigation of the bending stresses in the cylinders caused by elliptical distortion was made and this deformation was taken as being that which the nose-liner guides when set to their prescribed clearances would permit, and consequently this clearance of $^1/_4$ in. in diameter then became the greatest possible departure from a true circle. The formula for the maximum bending moment produced by this deformation was found to be:

$$M = QRu\left(\frac{1}{1 - (Q/Q_{cr})}\right)$$

where M = maximum bending moment, in-lb
Q = external pressure on gate, lb per sq in.
Q_{cr} = critical pressure, lb per sq in.
u = deformation, in.

It will be noticed that if the critical pressure is substituted for Q in this equation, the value of the moment then becomes infinite and the ring is beyond the point of stability.

Since each cylinder consists of a plate shell or skin plate with a number of internal stiffener rings welded thereto, it was obvious that local stresses would occur within the ring. These stresses were calculated from the following formula:

$$M = Ba^2Q$$

where M = moment, in-lb
B = 0.083
a = distance between stiffeners, in.
Q = external pressure, lb per sq in.

This is not strictly correct as the plate really consists of a circular shell and the modulus of rigidity should be involved. However, the error is on the side of safety. This stress acts at right angles to the other two. These stresses were summed to give the maximum resultant stress.

In completing the final checking of the designs it was found that when the upper gates are subjected to their maximum design pressure the external water load would cause them to contract 0.097 in. in diameter, while the lower gates when subjected to their maximum working pressure would contract 0.093 in. in diameter.

STEEL-PLATE SPECIFICATIONS

The newly revised designs having been completed, there still remained the selection of the rolled-steel plate which would most nearly satisfy all these conditions hereinbefore set forth, and after steel-mill representatives and authorities on welding were consulted on this subject the following specifications were prepared:

Specifications required that steel plates be made by the open-hearth process, that they should contain 0.2 to 0.3 per cent copper, and show a yield strength of not less than 24,000 lb per sq in. based on 0.2 per cent permanent elongation of original gage length. The plate material actually used had a yield point of 27,000 lb per sq in., an ultimate tensile strength of 58,500 lb per sq in., and an elongation in two inches of 36 per

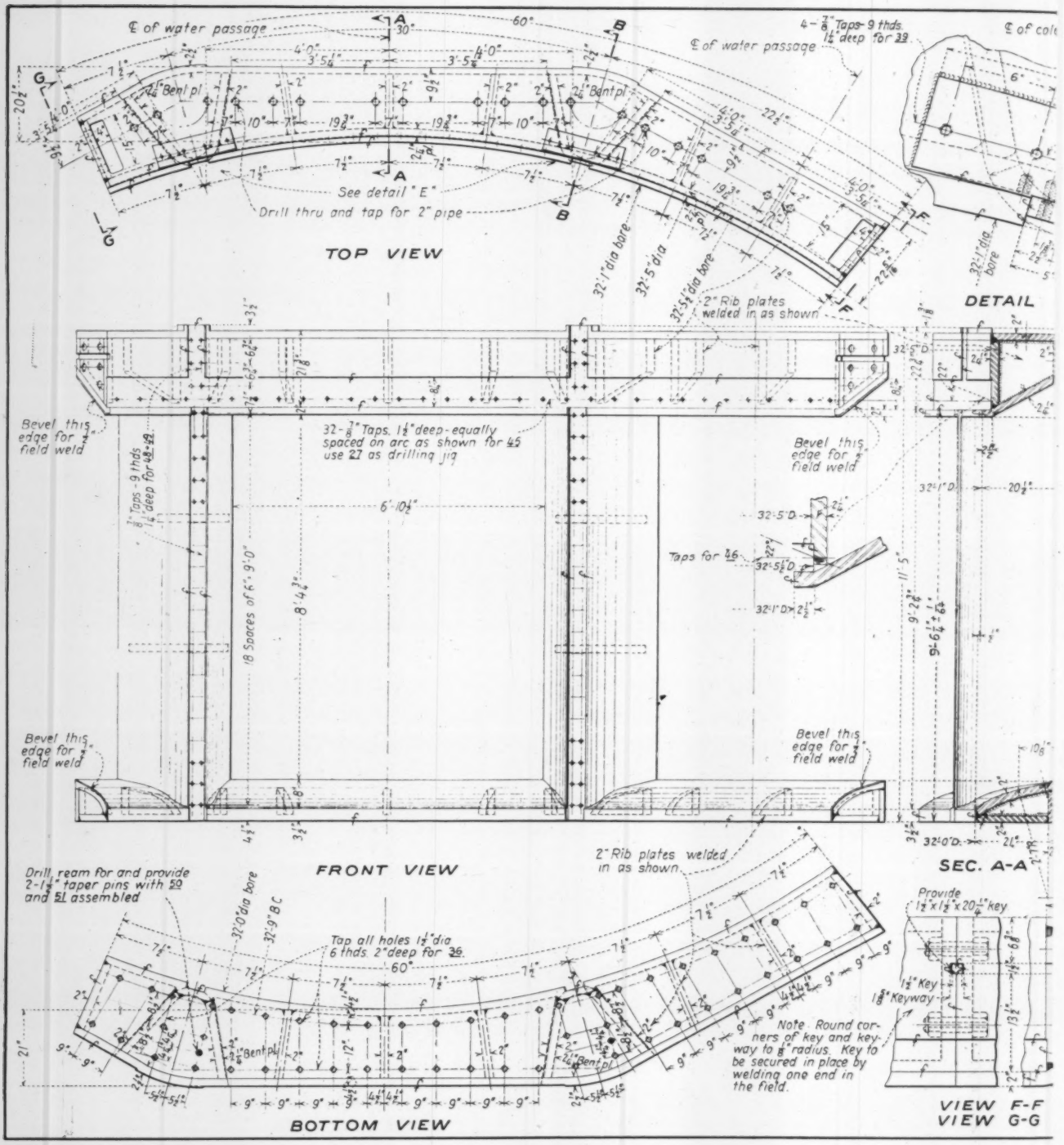

℄ of water passage
See detail "E"
Drill thru and tap for 2" pipe
TOP VIEW
℄ of water passage
4 - 7/8" Taps 9 thds. 1½" deep for 39
DETAIL
2" Rib plates welded in as shown
Bevel this edge for ¾" field weld
32 - 7/8" Taps. 1½" deep - equally spaced on arc as shown for 45 use 27 as drilling jig
18 Spaces of 6" = 9'-0"
Taps for 46
FRONT VIEW
SEC. A-A
Drill, ream for and provide 2 - 1¾" taper pins with 50 and 51 assembled
Tap all holes 1½" dia 6 thds. 2" deep for 36
2" Rib plates welded in as shown
Provide 1½" x 1½" x 20¼" key
1½" Key
1⅞" Keyway
Note. Round corners of key and keyway to ⅜" radius. Key to be secured in place by welding one end in the field.
BOTTOM VIEW
VIEW F-F
VIEW G-G

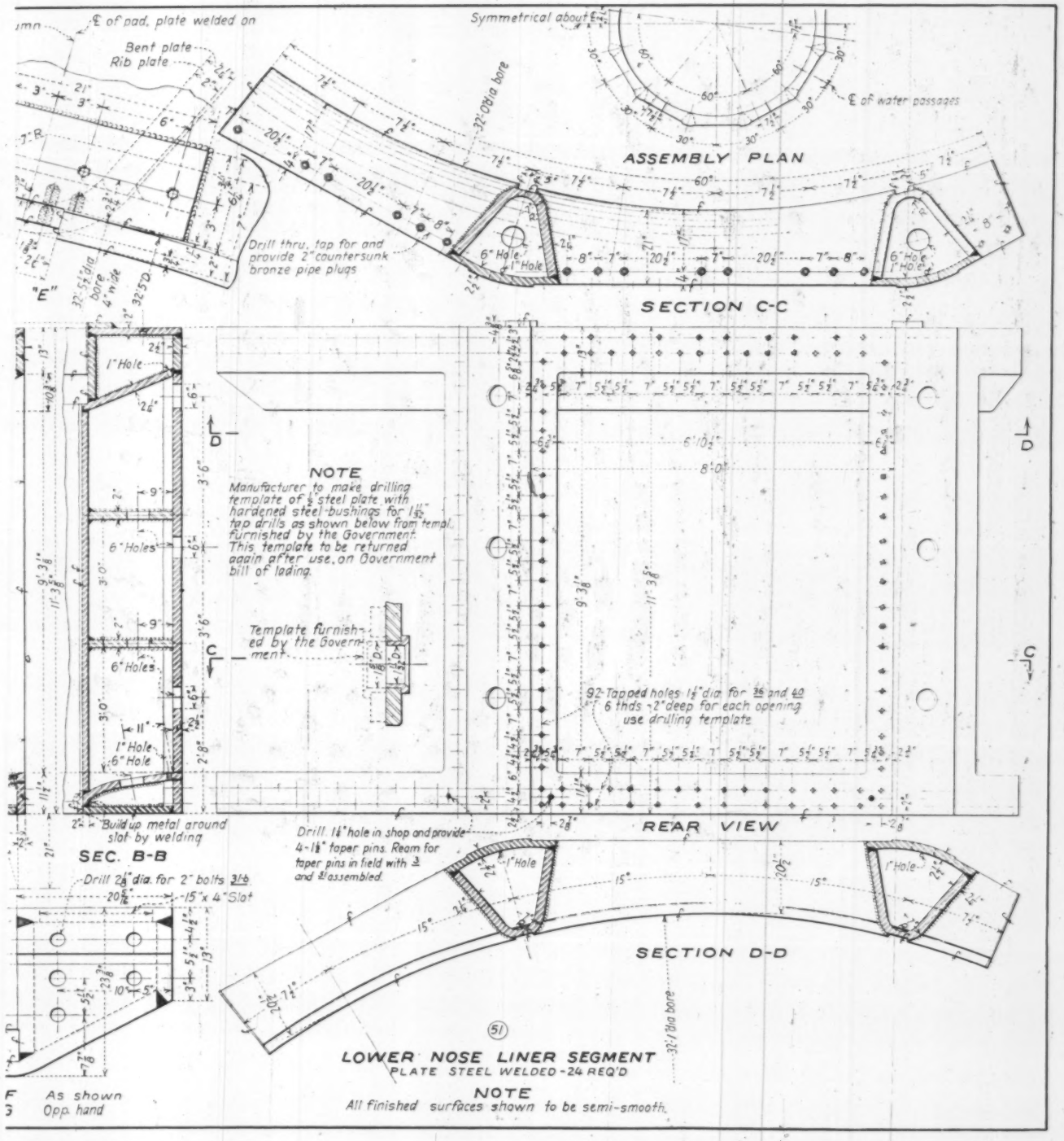
FIG. 5 LOWER

cent. The chemical analysis was approximately as follows: carbon 0.23, manganese 0.56, phosphorus 0.029, silica 0.029, and copper 0.24.

WELDING SPECIFICATIONS

Specifications required all welding to be done in the shop to conform with Par. U-69, of the A.S.M.E. Code for Unfired Pressure Vessels. Only skilled welders capable of meeting the test requirements for welders under the rules of the A.S.M.E. code could be employed on this work, and the welding process was required to be under control at all times.

Plates to be joined by welding were required to be accurately cut to size and shape with their edges so prepared as to allow thorough fusion and complete penetration on all joints. The surface of the plates for a distance of one-half inch back from the welding edge was required to be thoroughly cleaned of all rust, grease, and scale to bright metal. Flame cutting was permitted providing it was done by machine. Irregularities


ASSEMBLY PLAN
SECTION C-C
REAR VIEW
SEC. B-B
SECTION D-D
NOTE
Manufacturer to make drilling template of ½ steel plate with hardened steel bushings for 1½" tap drills as shown below from templ. furnished by the Government. This template to be returned again after use, on Government bill of lading.
Template furnished by the Government
92-Tapped holes 1½" dia. for 36 and 40 6 thds 2" deep for each opening use drilling template
Drill thru, tap for and provide 2" countersunk bronze pipe plugs
Drill 1¼" hole in shop and provide 4-1¼" taper pins. Ream for taper pins in field with 3 and 5 assembled.
Build up metal around slot by welding
Drill 2⅛" dia. for 2" bolts 3 I-b
15" x 4" Slot
(51)
LOWER NOSE LINER SEGMENT
PLATE STEEL WELDED - 24 REQ'D
NOTE
All finished surfaces shown to be semi-smooth.
F As shown
G Opp. hand

NOSE-LINER SEGMENT

of contours at critical stress points were required to be removed by welding and grinding.

Each layer of deposited weld metal was required to be thoroughly cleaned and peened before the next succeeding layer was deposited. The welding of all joints was required to be done by a process that would exclude the atmosphere from the metal while in its molten state. All finished welds were required to be reasonably smooth and free from grooves, depressions, undercuts, and other irregularities.

The dimensions of the welds were not given in the specifications but the contractor was required to proportion all welds so that after heat treatment the full strength of the connected members would be developed. It was required that the stress in the welded joints would not exceed 10,000 lb per sq in. in tension and compression, or 7500 lb per sq in. in shear.

ANNEALING AND STRESS RELIEVING

After the completion of all welding on each segment it was

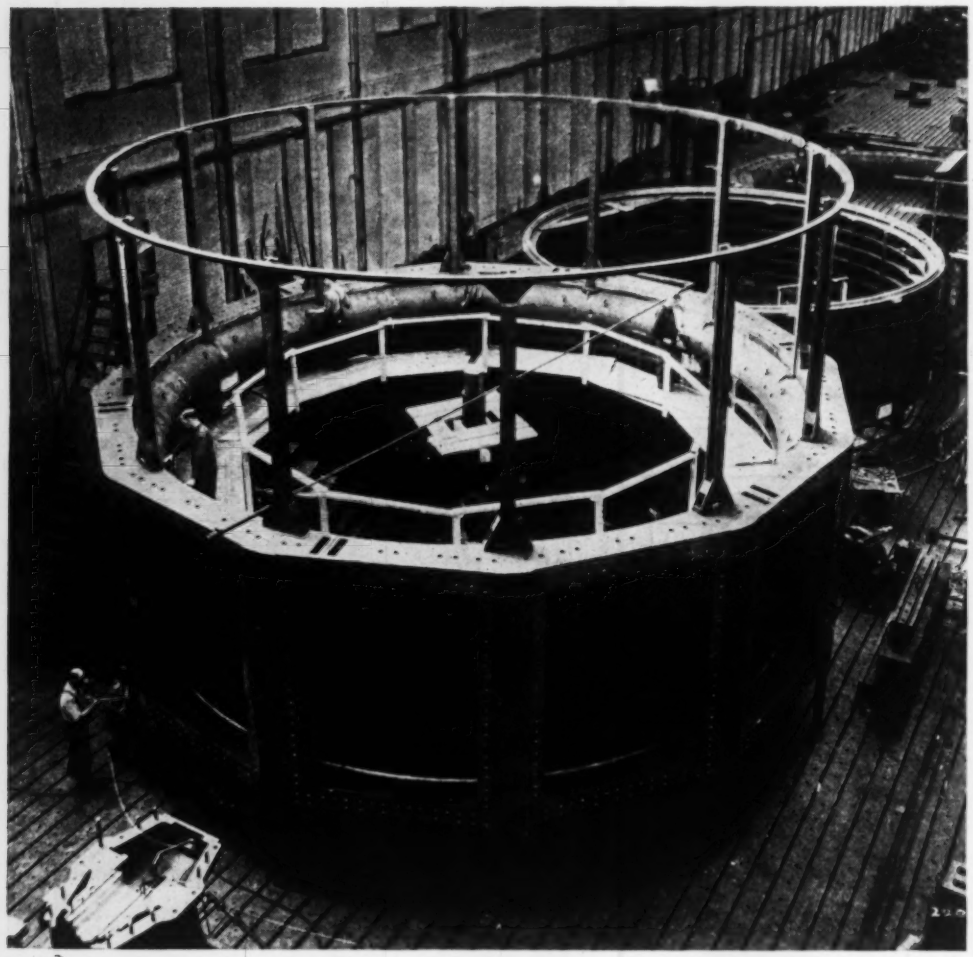


FIG. 6 SHOP ASSEMBLY OF FIRST GATE WITH DEFLECTORS, NOSE LINERS, AND UPPER GATE FRAME

(In middle distance is cylinder gate No. 2, while still farther back a throat-liner assembly is seen. The men in the foreground and on the temporary circular staging inside the gate afford an excellent means for comparison of size of these structures.)

required that the segment be annealed at proper temperature for a period of time not less than one hour for each inch of thickness of maximum metal section, and then allowed to cool slowly in the furnace. After heat-treating and machining were completed and before final inspection and painting, the outside unfinished surfaces were to be shot or sand blasted to bright metal.

FABRICATION OF STEEL-PLATE WORK

Government specifications No. 541 embodying the complete set of detail drawings for the cylinder gates, entrance liners, etc., were prepared and issued, and on opening bids on January 27, 1933, the Westinghouse Electric & Manufacturing Company was found to be the low bidder on the cylinder gates and nose liners, while the Goslin-Birmingham Manufacturing Company, Inc., was low bidder on the semi-steel entrance liners. After a brief interval contracts were awarded to these firms requiring delivery of the first lower cylinder gate and nose liners within 150 calendar days. The entrance liners for this first gate were also required within 150 calendar days, with heavy penalties for failure to make deliveries within the dates specified. The Westinghouse Electric & Manufacturing Company began work at once in their East Pittsburgh plant on the fixtures and jigs required for the fabrication of the cylinder gates, while in Birmingham, Ala., the Goslin-Birmingham Manufacturing Company, Inc., started the production of the 2,500,000 lb of semi-steel castings required for the entrance liners.

TRAINING THE WELDERS

An early survey of the amount of welding involved revealed that 530,448 linear inches of welding was required, or substantially 8.4 miles, and it was evident that an adequate force of welders should be made available so that delivery could be assured within the limited time allowed. The Westinghouse shops accordingly became active in the training of welders who could qualify under the A.S.M.E. Code.

In all about eighty qualified welders were finally engaged in the construction of the cylinder gates and nose liners, and the course of training employed to enable them to pass the necessary tests and qualify under the A.S.M.E. Code requirements so that they could actually weld on the job was substantially as follows:

A day was devoted to special instruction and demonstration to inform the welders of the requirements and quality of the welds and of the proper welding technique to be employed in producing the welds. This was followed by from five to ten days of practise in making welds under the supervision of an instructor. The length of time devoted to this practise period varied with the ability and aptitude of the men receiving the instruction. During this period the welders were engaged in

making butt welds between plates about 15 in. long with thicknesses ranging from $1^1/_2$ in. up to 3 in., which is the maximum section used in these gates. After completion these welds were subjected to nick-break tests so that the fractured metal could be examined and the proficiency of the welders determined.

When a welder was finally able to produce welds which, when nick-break fractured, showed no inclusions of slag and were free of gas holes and poor fusion, he was then considered to be capable of making welds with high-quality welding electrodes such as those used to produce welds which would meet the requirements for tensile strength, ductility, and soundness as specified in the A.S.M.E. Code. Three typical nick-break fractures made by welders trained in this manner are shown in Fig. 8.

Included in the practise period was training in the making of single- and double-fillet welds.

In those instances where welders failed to pass their final tests, the reasons for their failures were explained to them and they were then given special instructions on how to correct their faults, after which they were required to practise for several days before again being permitted to make another test plate.

QUALITY OF WELDS

The A.S.M.E. Code for Class 2 welding requires that all welds be made with a high-quality fluxed electrode so that the tensile strength as determined by full-section tensile specimens will not be less than the minimum specified tensile range of the plate, and the tensile strength by reduced-section tensile specimen not less than 95 per cent of the minimum specified tensile range of the plate; ductility by free bend, 20 per cent elongation, and nick-break fracture to show sound, clean metal with not more than six gas pockets $^1/_{16}$ in. maximum diameter per square inch of weld surface exposed in the fracture. To fulfil these requirements the electrodes used throughout the entire job were of the Westinghouse "crucible weld" type, and the quality of the welds made by these electrodes actually met all the physical requirements of Paragraph U-68 of the A.S.M.E. Code for Unfired Pressure Vessels.

Specimens made by the welders produced the following typical results:

Full-Section Tensile. Failure in the parent metal at an ultimate strength equal to the parent metal (55,000 to 65,000 lb per sq in.).

Reduced-Section Tensile. Failure in the parent metal at an ultimate strength equal to that of the parent metal (55,000 to 65,000 lb per sq in.).

Ductility. 30 to 50 per cent by free bend.

Nick-Break. Maximum of one to two holes less than $^1/_{16}$ in. in diameter in a specimen with $4^1/_2$ sq in. of exposed surface in the fracture.

WELDING TECHNIQUE

The diameters of the electrodes used were $^3/_{16}$, $^1/_4$, and $^5/_{16}$ in. The $^3/_{16}$-in. electrodes were used primarily for tacking and

FIG. 8 NICK-BREAK FRACTURES OF WELDED SPECIMENS

for depositing the first beads in the bottom of butt welds where the gap was too narrow to obtain good penetration with $^1/_4$-in. electrodes. Electrodes $^1/_4$ and $^5/_{16}$ in. in diameter, depending upon the size and location of the weld, were used to finish the butt welds. All fillet welds were made with $^1/_4$-in. electrodes.

The design of the cylinder gates and nose liners was such that the fusion welding of the main elements in the butt welds could be accomplished in most instances with single bevel-type joints. A small percentage of the joints was made of the single- and

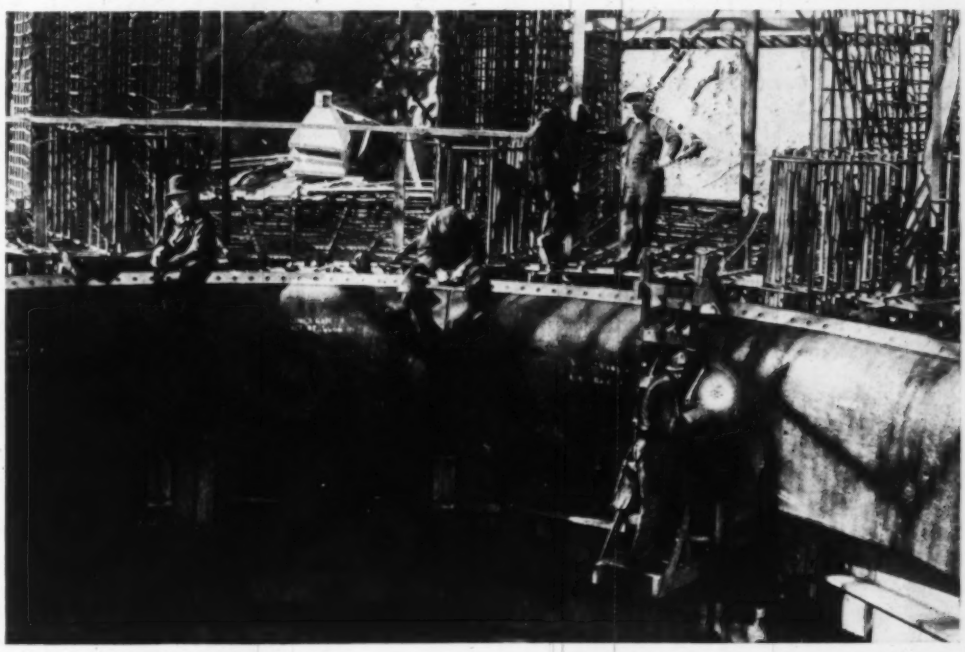


FIG. 7 FIELD WELDING SEGMENT JOINTS OF CAST-STEEL THROAT LINER, SUPPORTED BY STRUCTURAL-STEEL CORBELS PROJECTING FROM WALLS OF CONCRETE-LINED SHAFT

(Basket weave of reinforcement bars with gaps for entrance-liner castings may be seen in background.)

FIG. 9 FILLET WELDING VERTICAL SPACERS TO HORIZONTAL REINFORCING-RING RIBS FOR CYLINDER-GATE SEGMENT

double-vee types. The included angle of the kerfs of the single-bevel joints was held as closely as possible to 45 deg and the included angle between the kerfs of the single, and double-vee joints was held to about 40 deg. At first thought these angles might seem small, but extensive tests and investigations revealed that they gave the best results. The only difficulty that might be encountered in joints of this type with such a small included angle would be in obtaining good penetration at the bottom and this difficulty was completely eliminated by slightly spacing the butting edges wherever possible and/or by depositing the first beads with small ($^3/_{16}$-in. diameter) electrodes.

The advantages of the small included angles of the kerfs were found to be as follows:

(1) It reduced the amount of weld metal required in making a joint, thereby reducing the welding time and the cost.

(2) The reduced amount of deposited metal reduced the local heating in the structure and reduced the residual stresses.

(3) It cut down the maximum width of the joint and made it possible to weld the full width of the widest joints at a single pass without excessive weaving. It has been found that if a very wide weave is used with fluxed electrodes the pool of molten metal becomes too large, thereby resulting in overheating of the deposited weld metal and excessive undercutting.

OPERATIONS ON THE CYLINDER GATES

Inasmuch as the upper and lower cylinder-gate designs are essentially the same, the same general welding procedure could be used on both; hence only the work on the lower gates, which were fabricated first, will be described.

The Westinghouse crucible-weld type electrodes are of the fluxed type, primarily adapted for horizontal welding. Because of this fact and because larger electrodes could be used and greater welding speeds obtained by horizontal welding, the parts being joined by the welding processes, including each fully assembled gate segment, were tilted in various positions during the welding process so that all the joints could be welded in that position. When making fillet welds the structures were tilted approximately 45 deg in order to form a trough and so simulate a butt weld.

Tilting of the completely assembled gate segments into the many necessary welding positions was a large task, consequently all parts that could be individually assembled and welded before the complete assembling of the segments were so made. This procedure made possible the use of simple fixtures such as those shown in Fig. 9, where the welders are shown making fillet welds to join the vertical spacers to the horizontal reinforcing ribs of a gate segment. The simplicity of the jigs and the relatively small sizes of the parts welded in such cases aided greatly in increasing welding speeds.

After welding the necessary individual details, the complete segments, with the exception of the parting plates which are attached to the ends of the reinforcing ribs to form the radial-joint flange for bolting to the adjacent section in final assembly,

FIG. 10 PARTIALLY ASSEMBLED GATE SEGMENT SHOWING LOCATION OF $^1/_2$-IN. TACK-WELDS

(Operator is just starting final welding in center of segment.)

were assembled in a jig and tack-welded together. These tack welds were 1/2-in. welds approximately 6 in. long. Their locations are clearly shown in Fig. 10.

After the gate segments were tack-welded together they were removed from the assembly jig and the reinforcing ribs, the rib spacers, and the shell plate were completely welded together. This being completed, the ends of the reinforcing ribs were machined and beveled and the parting plates forming the radial flanged joints at the ends of the segments were attached and butt-welded into position.

Fig. 10 shows a gate segment tilted at approximately 45 deg for making the fillet welds between the reinforcing ribs and the shell plate. These welds were made in short lengths (about 8 in. long) for the full thickness of the finished weld, working progressively from the center of the segment toward the outer edges. Three layers made with 1/4-in. electrodes were used in these welds. The purpose of this procedure was to prevent as much as possible the presence of any welds smaller than the finished size. Making welds by depositing single layers the full length of the joints was not practicable because the uneven heating and cooling of the thick heavy members would have caused them to crack. Constructing the welds so that all but the immediate section being welded were of full size eliminated this difficulty.

Butt welds were made by progressing from the center of the joint toward the ends.

In order properly to weld the rib spacers and parting plates in place, the segments were mounted on a fixture as shown in Fig. 11. This fixture was constructed so that the segments could be rotated and tilted inward or outward. This made it possible to place the segment at any desired angle or position for welding.

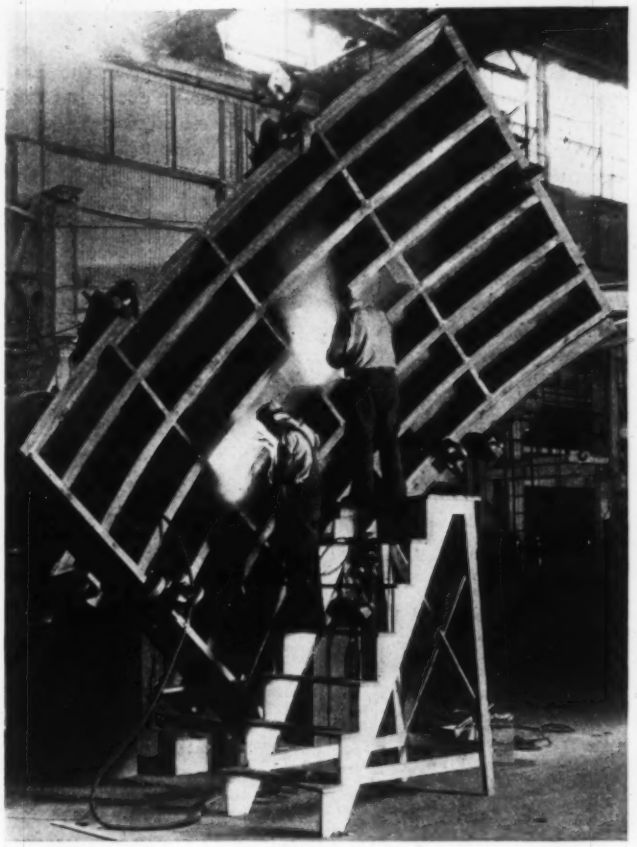


FIG. 11 UNIVERSAL TILTING AND ROTATING WELDING FIXTURE WITH COMPLETELY ASSEMBLED GATE SEGMENT IN PLACE AND OPERATORS FORMING TWO FILLET WELDS BETWEEN THE REINFORCING RING RIBS AND THE VERTICAL SPACERS

(The segment shown in the fixture weighs 31,600 lb.)

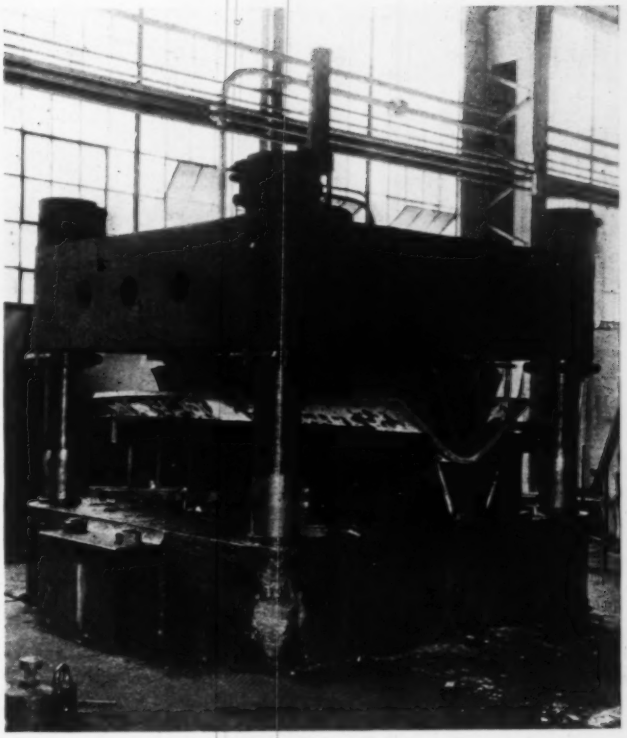


FIG. 12 HEATED NOSE-LINER PLATE BEING FORMED IN SPECIAL DIES IN 1500-TON HYDRAULIC PRESS

In addition to the procedure already described the following general rules were followed:

(1) All layers of deposited weld metal were peened with a blunt-nosed peening tool operated by a light chipping hammer in order to reduce the residual stresses and clean the weld. The peening operation was carried out immediately after the deposition of each six to eight inches of deposited metal. Peening in this manner before the deposited metal became cold gave better results.

(2) After welding was once started on a gate segment it was continued, as nearly as possible, to completion without interruption. This procedure prevented the segments from being successively heated and cooled and consequently reduced the thermal strains.

(3) All tack welds were chipped out during the main welding process. Although this procedure was not necessary in all cases it was found to produce the best results because of the danger of welding over a tack that contained an initial crack.

(4) Special attention was given by each welding operator to all craters. For the best results it was found desirable to fill all craters before breaking the arc. This was accomplished by lengthening the arc slowly before breaking it.

STAINLESS-STEEL BARS NOT WELDED

As embodied within the original government specifications, 5/8-in. by 4-in. stainless-steel bars 9 ft 1/2 in. long were to be inserted into vertical machined slots at 30-deg intervals around the outer circumference of each gate to register opposite the inlayed bronze guide bars in the end faces of the nose liners and

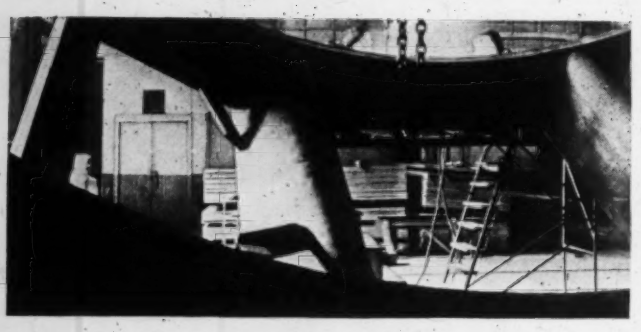


FIG. 13 TILTED NOSE-LINER SEGMENT HELD IN DESIRED POSITION FOR WELDING BY OVERHEAD CRANE

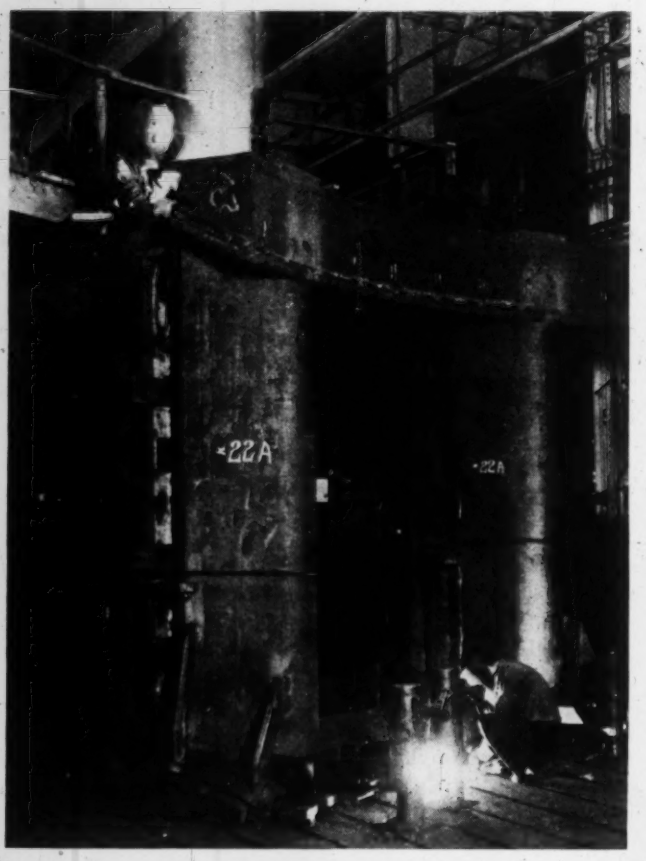


FIG. 14 COMPLETELY ASSEMBLED NOSE-LINER SEGMENT RIGIDLY CLAMPED IN MAIN WELDING JIG WHILE FINAL WELDING IS DONE
(The temporary tack-welded braces are to prevent distortion while welding is being done.)

the extensions of the bars in the columns of the gate frames above. These stainless-steel bars were to be welded into the gate along each edge, as shown in detail *F* of Fig. 4.

When the Westinghouse technical staff investigated this requirement they recommended that these guide bars be attached with screws instead of welding, giving the following reasons:

The presence of chromium in steel lowers the carbon content in the microstructural constituent pearlite, raises the temperature at which the carbon change point occurs (AC–1 point), decreases the speed at which carbon diffuses through the heated steel, and produces the property of air hardening. The most important of these effects from the standpoint of welding is that of air hardening.

The amount of chromium present in the stainless steel specified for the guide bars would give a composition of carbon 0.12 per cent, manganese 0.50 per cent, silica 0.50 per cent, phosphorous 0.03 per cent, sulphur 0.03 per cent, and chromium 11 to 15 per cent, which is sufficient to make the steel air-hardening. In other words, the guide bars would harden on air cooling from temperatures above the AC–1 point. Since in welding these bars the metal a short distance from the weld would be heated above the AC–1 point, it would, as a result, become hardened during the cooling process. This hardening would actually be more severe than simple air hardening because of the large cooling masses of adjacent metal in the gates. As a result the stainless steel bars would contain areas with an austenitic and/or martensitic structure. These areas would be extremely hard and brittle and susceptible to failure resulting from welding strains or sudden impact loadings.

Although it would be possible to weld straight 11- to 15-per cent chrome steel by the use of ductile welding electrodes, it would not be possible to insure sound strong joints that would stand up in service. Even though cracks would not be apparent in or adjacent to the welds, it would be highly probable that the austenitic and/or martensitic structures adjacent to the welds would contain small or microscopic cracks resulting from the stresses set up from the welds in cooling. Such microscopic cracks might develop in service and ultimately cause failure.

The stainless-steel guide bars, accordingly, were fastened to the gate shells with screws, as recommended.

OPERATIONS ON THE NOSE LINERS

The nose liners, as shown in Fig. 5, were made in the same general manner as the cylinder gates, with the exception that the vertical wedge-shaped nose plates were heated and formed between special die blocks in a 1500-ton hydraulic press. The beginning of this forming process is shown in Fig. 12. The top and bottom ring plates were likewise formed in this same general manner. Wherever possible, individual sections such as the noses and top ring, were separately welded before being introduced into the general assembly. The general assembly was accomplished by placing the various parts in a jig and tack-welding them together. During this assembling process numerous braces were added between the members to prevent distortion. Fig. 13 shows the location of some of the braces applied.

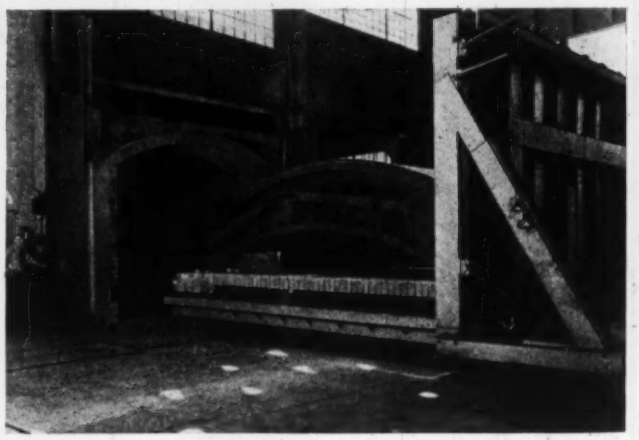


FIG. 15 NOSE-LINER SEGMENT AND CYLINDER-GATE SEGMENT ON SPECIAL CAR ENTERING ANNEALING FURNACE ARCHWAY AT LEFT
(Inside the archway some of the electric heating elements may be seen on the furnace wall.)

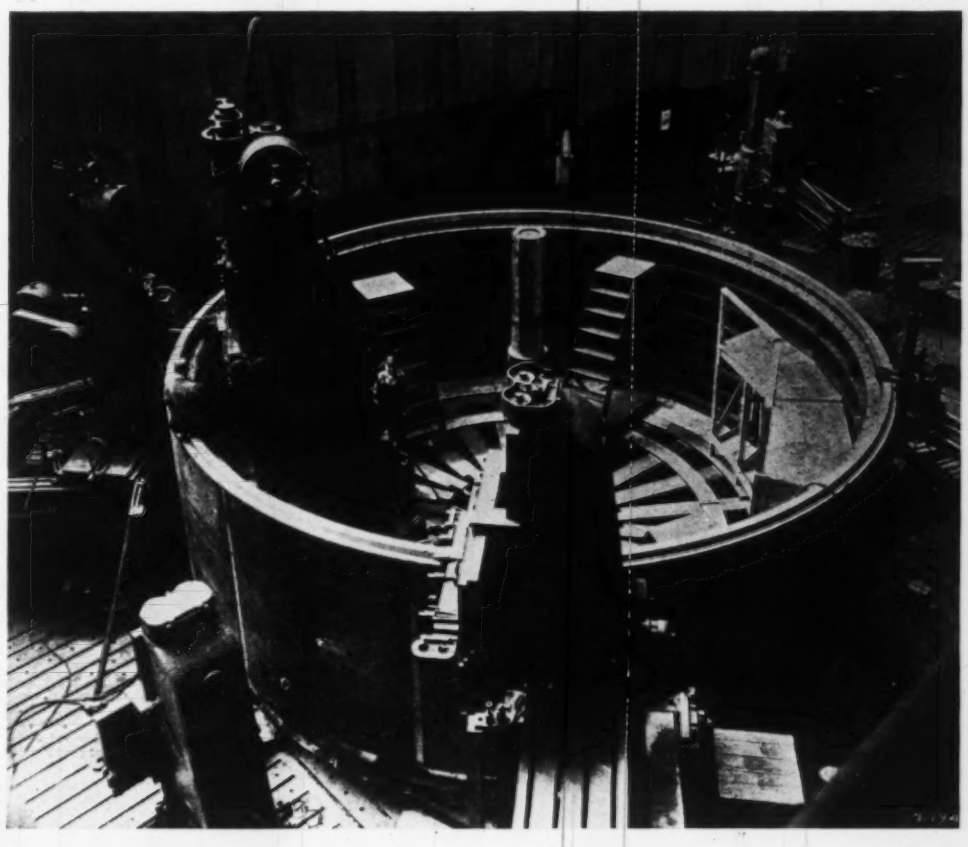


FIG. 16 ASSEMBLED CYLINDER GATE BEING BORED AND TURNED SIMULTANEOUSLY

(A vertical slotter and a special slotting and drilling machine have also been moved up and are ready to go to work.)

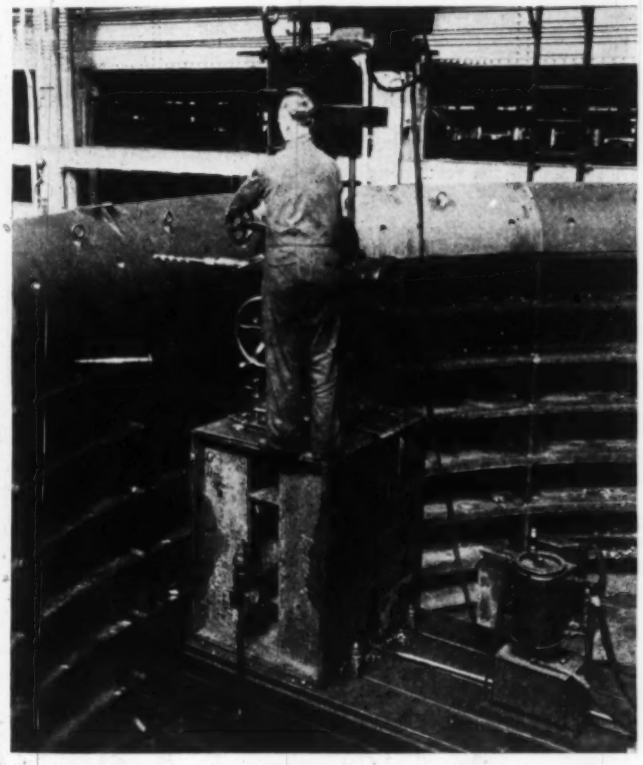


FIG. 17 SPECIAL MACHINE SET-UP INSIDE CYLINDER GATE DRILLING DEFLECTORS

In order that the nose-liner segments might be tipped so that all welding could be done in a horizontal plane, they were suspended by means of overhead cranes as is shown in Fig. 13. This afforded a quick cheap means of support and turned out to be highly satisfactory.

In Fig. 14, the final welding of the completely assembled nose-liner segment is shown. In the background and to the left can be seen the heavy welded structural-steel main assembly jig used to maintain the units comprising each nose-liner segment in proper positions and to prevent distortion during welding. Some of the temporary tack-welded braces can be seen in the lower foreground, also the hydraulic pull-down jacks. A rolled-steel heavy-section H-column, tack-welded in as a temporary vertical spacer, may be seen at the right between the end extremities of the segment. These segments weigh 30,500 lb each.

The same general rules and methods for making butt and fillet welds as outlined for the cylinder gates were followed in the fabrication of the nose liners.

ANNEALING THE WELDED PARTS

After completing the welding of each gate or nose-liner segment it was strain-relieved in an electrically heated annealing furnace at a temperature of approximately 1300 F. The annealing cycle consisted of 18 hr to bring the furnace up to temperature, 3 hr soaking time at the required temperature, and 14 hr to cool the furnace to 100 F before removing the parts. Fig. 15 shows a nose-liner segment with a gate segment resting upon it, loaded upon an oven car ready to be moved into the annealing furnace.

During the welding processes the gates and nose liners had a tendency to warp slightly out of shape. This condition was effectively corrected during the annealing process by placing weights on top of the segments and supports underneath them. The proper location of the weights with respect to the supports caused the segments to bend back into their correct shapes during the annealing process, and the maximum out-of-round

of any of the finished gate segments was held within plus or minus $^1/_4$ in. on the 32-ft diameters.

MACHINING OPERATIONS

An excellent idea of the heavy equipment used in performing some of the major machining operations may be gained from Fig. 16, which shows one of the gate assemblies being bored and turned simultaneously. A vertical slotter and also a special slotting and drilling machine have been moved up and are ready to go to work.

Fig. 17 shows an interior view of one of the lower gates with a special drill set-up at work on the deflectors on top of the gate.

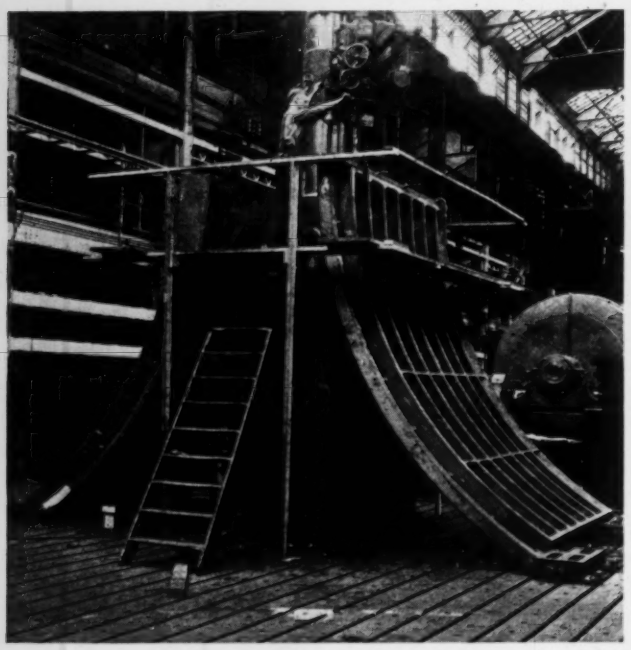


FIG. 18 DRILLING BOLT HOLES IN RADIAL PARTING FLANGES

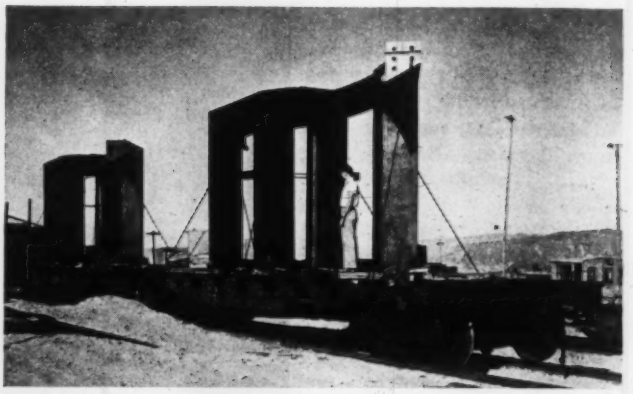


FIG. 20 NOSE-LINER SEGMENTS ON CARS IN FREIGHT YARDS

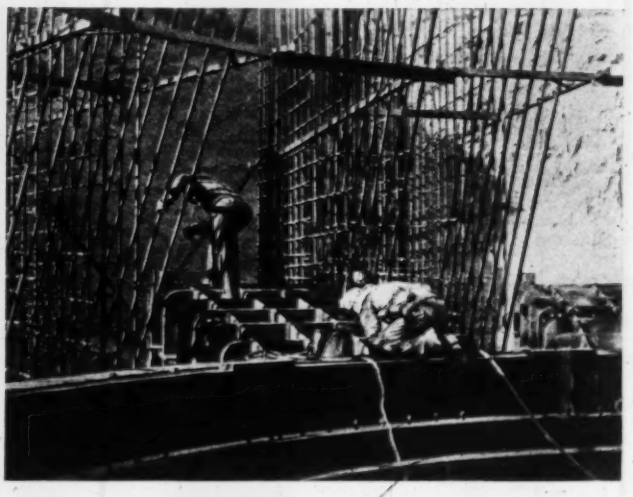


FIG. 21 INSTALLING ENTRANCE-LINER CASTINGS IN FIELD
(A portion of the reinforcing steel for two of the radial buttress-tower walls is seen.)

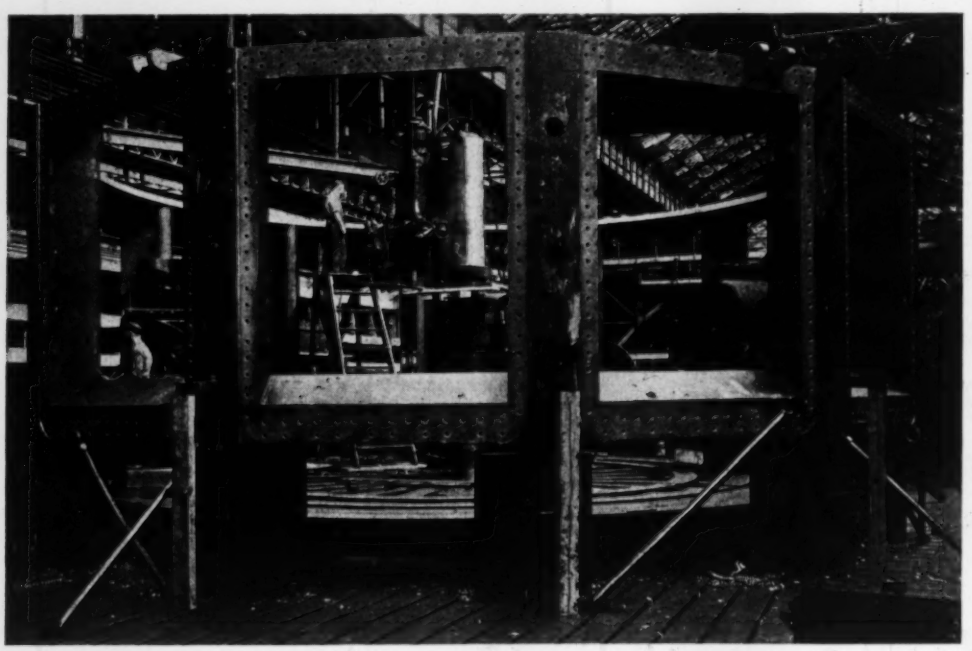


FIG. 19 BORING NOSE-LINER ASSEMBLY

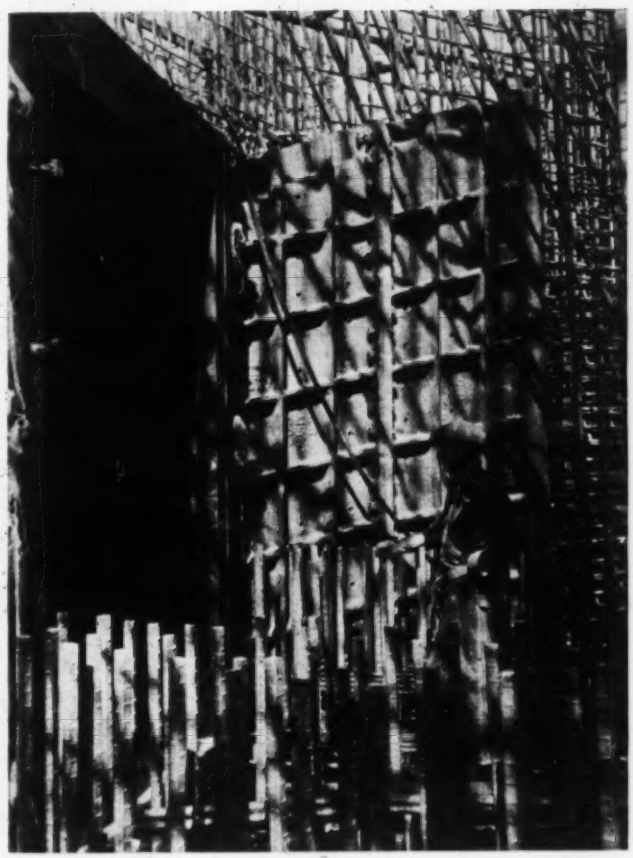


FIG. 22 OPENING ON LEFT IS ONE OF THE NOSE-LINER WATER PASSAGES

(An entrance-liner casting will be installed and bolted to the flange face in similar manner to that shown beyond the workman. Network of reinforcing bars makes a veritable jungle of steel.)

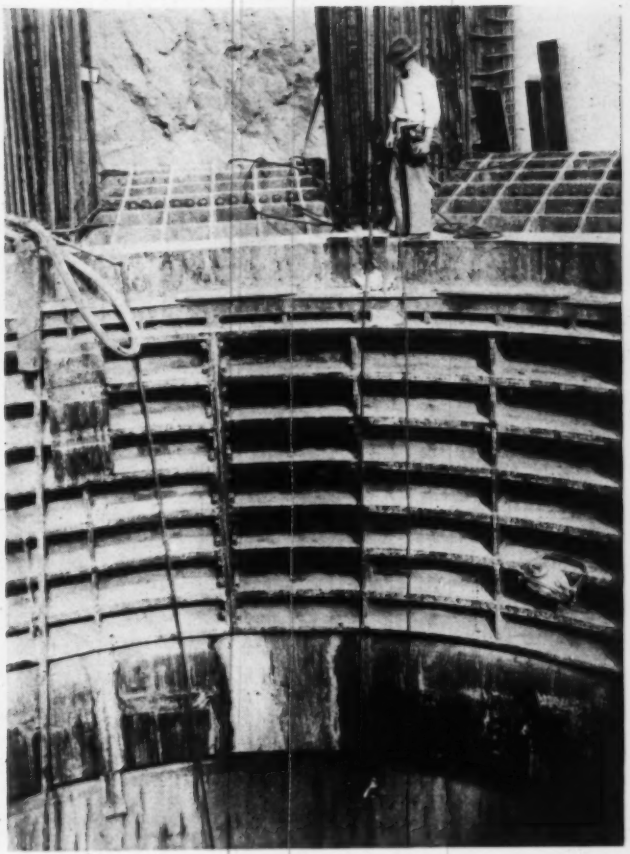


FIG. 23 INTERIOR VIEW OF A BOTTOM CYLINDER GATE IN CLOSED POSITION

(Tops of two entrance liners are visible, while below the throat liner is seen, carried on corbels, ready for the upper end of one of the 30 ft steel outlet pipes to be connected.)

FIG. 23*a* LOOKING DOWN ON DOWNSTREAM NEVADA INTAKE TOWER, WITH TOP OF DAM STRUCTURE IN CENTER, AND THE CURVING UPSTREAM FACE OF THE DAM IN LEFT CENTER

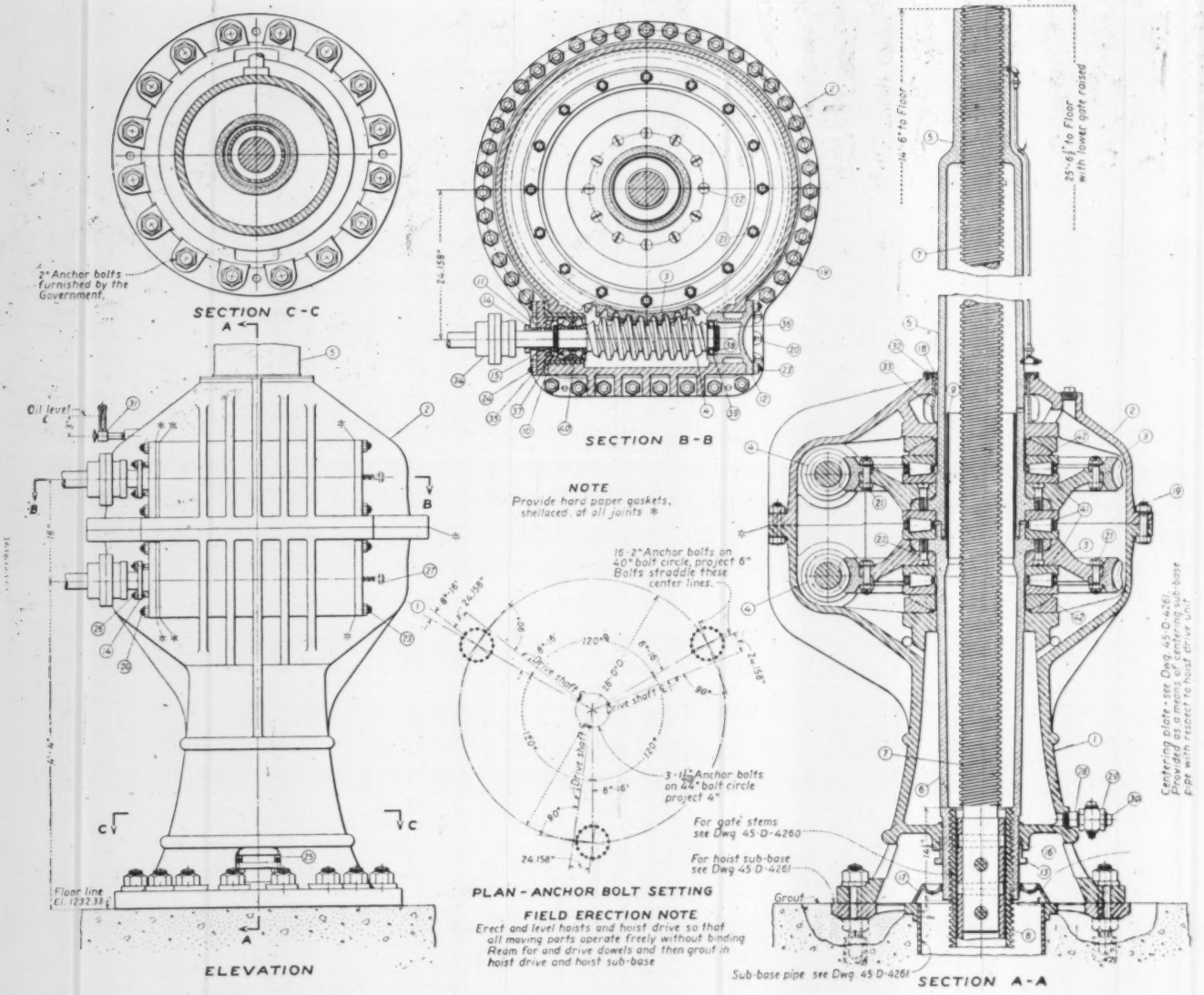

2" Anchor bolts furnished by the Government.
SECTION C-C
SECTION B-B
NOTE
Provide hard paper gaskets, shellaced, at all joints *
Oil level
Floor line El. 1232.33
ELEVATION
16-2" Anchor bolts on 40" bolt circle, project 6" Bolts straddle these center lines.
3-1½" Anchor bolts on 44" bolt circle project 4"
For gate stems see Dwg 45-D-4260
For hoist sub-base see Dwg 45-D-4261
PLAN - ANCHOR BOLT SETTING
FIELD ERECTION NOTE
Erect and level hoists and hoist drive so that all moving parts operate freely without binding. Ream for and drive dowels and then grout in hoist drive and hoist sub-base
14'-6" to Floor
25'-6½" to Floor with lower gate raised
Grout
Sub-base pipe see Dwg 45-D-4261
SECTION A-A

FIG. 24 ASSEMBLY OF CYLINDER-GATE HOIST

Fig. 18 shows two gate segments upended and clamped on opposite sides of a special pedestal fixture on which is mounted a radial drill press used for drilling the bolt holes in the parting flanges of the segments.

Fig. 19 shows a completely assembled nose liner, turned bottom face up, being bored to receive the bottom seat.

The tolerances in the machining of these gates were held to the same precise limits as those employed in the fabrication of electric generators, and the fidelity with which the finishing of the monel metal seats, and in fact all the machining, was done, may be gained from the fact that the government inspector, F. W. Haversack, repeatedly commented upon the thoroughness and excellence of the work and in his acceptance report stated that the fully assembled gates in their closed positions successfully held water entirely around both their upper and their lower seals without leakage.

ACKNOWLEDGMENTS

The author takes this opportunity to thank Mr. Chas. H. Jennings, of the Westinghouse Research Laboratories at East Pittsburgh, Pa., for his kindness in permitting him to use much of Mr. Jennings' Science Paper No. 718 on welding in this article; and Mr. Lucian M. Cargo, executive representative for the Westinghouse Electric & Manufacturing Company in Denver, whose kind cooperation in securing the supporting data is much appreciated. Both the welding and machining phases of construction were performed under the able direction of Mr. C. C. Brinton, superintendent, Generator Division of the Westinghouse Company.

ERECTING THE GATES IN THE TOWERS

In shipping to the project, two nose liners were loaded on a flat car and clamped down with elaborate blocking and were braced from above by heavy steel rods, which may be seen in Fig. 20 where four segments are shown on the cars in the Boulder City freight yards. Figs. 21, 22, 23, and 23*a* which follow were taken as the gates were being installed.

In erecting the lower cylinder gates in their final positions in the intake towers, it was necessary first to prepare a heavily reinforced concrete slab ring about 98 ft outside diameter by 37 ft inside diameter by 10 ft thick, with 12 equally spaced I-beams placed radially in a common horizontal plane about midway of the slab's depth and extending inward toward the center of the shaft with their inner ends protruding as cantilevers from the inner ring face of the concrete and terminating in a circle approximately 31 ft in diameter, thus forming a set of

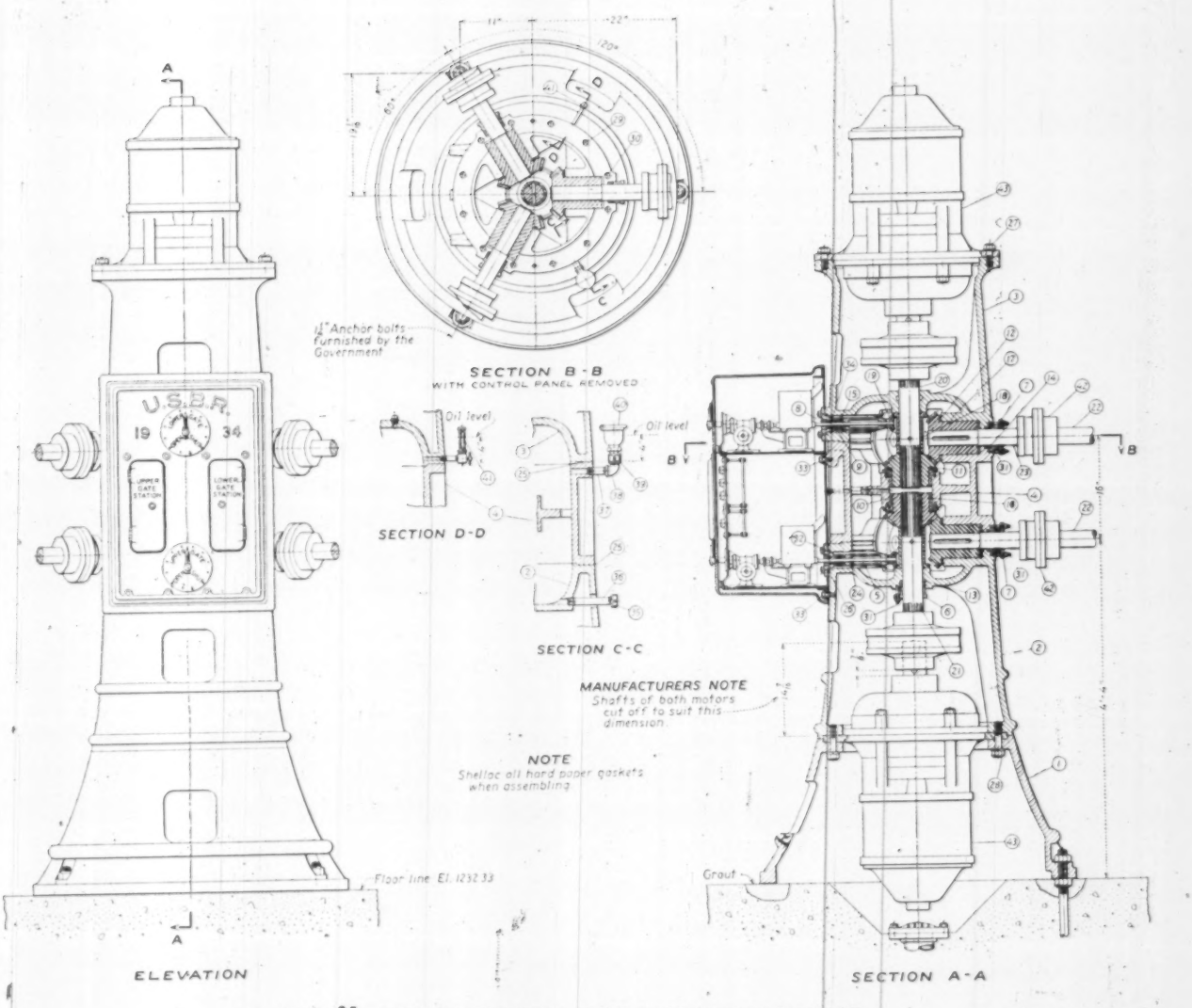


FIG. 25 ASSEMBLY OF CENTER DRIVE OF CYLINDER-GATE HOIST

heavy corbels upon which the throat liners of the lower gates were then assembled and leveled by means of jacks.

This arrangement was necessary in order to provide the working clearance required later to make the connection between the upper end of each 30-ft penstock header and the lower end of its complementary lower throat liner. After the gates were completely installed and the heavy rings of reinforcing steel and the concrete had been placed in the area above that where this connection between the throat liner and the pipe was to be made, the first section of pipe was pulled up from below and pinned and welded in place.

After the gates had been completely assembled and embedded in concrete these corbels were burned off so that the annular space around the lower portions of the throat liners was left in the clear.

The corbels are shown with the lower throat liner assembled and carried upon them in Fig. 7.

Because of the large size of the gates and the relatively rapid changes in temperature which occurred while they were being installed when subjected to the powerful sun rays in the arid country at the dam site and then to the cooling off which occurred when the canyon walls shaded them, some difficulty was experienced in assembling the seals into their proper locations. This trouble was found to be due to the difference of expansion coefficients of the monel-metal seals and the structural steel of the gates and was overcome by the use of canvas canopies to shut out the sun.

In Fig. 21, a near view of the upper edge of a bottom gate in the closed position is shown with the top inner faces of the nose liners. The tops of two entrance liners appear beyond, while the maze of reinforcing bars rises vertically and obliquely above.

Fig. 22 is an excellent close-up of one of the outer nose-liner flanges ready to receive an entrance-liner casting similar to that already bolted in place to the nose-liner face beyond. In the foreground the reinforcing bars are shown ready to receive the next vertical lift of bars 57 ft long, and in the right background the maze of reinforcing bars shows clearly.

In Fig. 23 a bottom cylinder gate is shown complete except for its rib-liner plate and the deflector castings. The gate is in the closed position resting on its bottom seat in the throat liner, which hangs out from the concrete shaft walls on corbels to provide the working clearance needed to install later the upper end of the 30-ft outlet pipe which will then be connected to it and will continue downward through the shaft to the tunnel below. In this view one of the radial-flanged bolted joints between two gate segments is shown.

In Fig. 23*a* the downstream Nevada intake tower is seen with concrete poured to about 60 ft above the foundations and the web-like weave of vertical reinforcing bars extending some 50

ft above the panel forms and the boom of the erection derrick beyond. Just above the right-hand corner of the square scaffold over the center of the intake tower a workman can be seen clinging to the vertical bars of reinforcing steel. The upper surfaces of the 60- by 50-ft rectangular concrete blocks in the dam show in the central portion of this illustration, the top of the highest concrete in the middle portion of the dam being at elevation 825 and 322 ft above the lowest portion in the river gorge which can be seen in the lower left-hand corner adjacent to the curving upstream face of the dam. Just to the left of the center of the intake tower six men can be seen at work leveling freshly dumped concrete in the raised 60-ft-square block, and further to the left the 8-cu yd concrete bucket which has just discharged its 25-ton burden appears as a very small tin can. By looking very closely many men can be discerned busily engaged in erecting forms, placing grout pipes, and doing many other tasks at numerous places on the dam, which, when completed, will rise 407 ft above the present level of the concrete shown in this picture, or 729 ft above the lowest portion of its foundations in the river gorge below.

CYLINDER-GATE HOISTS

In the intake towers above the gates at elevation 1232 are the operating floors, level with the roadway across the top of the dam, reached from the dam by means of concrete-masked steel-girder bridges of 107 ft span, as may be seen in Figs. 1 and 1*a*. On these floors are mounted the electric-driven triple-tandem hoisting heads which, through the agency of the three downwardly extending stems connected to each cylinder gate below, raise or lower the upper or lower or both gates as occasion may dictate. The general arrangement of these hoists may be seen in Figs. 1*a*, 24, and 25.

The stems connected to each upper gate are 187 ft long and the weight of the three is 40,100 lb, while the stems to the lower gate are 337 ft long and a set of three weighs 95,300 lb. The weight of moving parts for each top gate is 220,000 lb, and for each bottom gate it is 343,100 lb. Hoisting capacities of 365,000 and 500,000 lb, respectively, are being provided.

The stems connecting with the upper gates are tubular and are engaged by the lower portions of the hoisting heads and operated thereby, while the stems from the lower gates extend upward through the tubular stems of the top gates and are engaged and operated by the upper part of the hoisting heads. In Fig. 24, Section *A-A*, the relationship of the upper and lower gate stems with their hoisting mechanisms may be seen.

The upper portions of the three hoisting heads which operate the stems extending to the lower gate are interconnected with their own electric gear-motor drive in the center drive unit by means of horizontal shafts extending from the upper worm and gear reduction (Fig. 24, Section *B-B*) in each head to the center-drive speed-reducer unit so that they always operate in synchronism. A similar and independent center-drive motor with radial horizontal shafts drives the lower worm and gear-reducing mechanism in the lower portion of each hoisting head which operates the stems of the upper gate. By this arrangement either the upper or lower cylinder gates can be opened or closed independently or simultaneously. The relative arrangement of the center-drive unit with its three sets of radial drive shafts extending to the three tandem hoisting heads is shown diagrammatically in plan in Fig. 24, Plan of Anchor-Bolt Setting.

Control circuits are provided whereby emergency closure of both gates in any or all towers can be accomplished by remote electric control from the power house, but all other operations will be by direct control from the operating room in each tower. The control panel with its push buttons, indicator dials, limit switches, etc., is mounted on the center drive unit (Fig. 25).

The normal travel of the gates for opening or closing is approximately 9 ft, but the hoists are arranged so that whenever inspection or maintenance so requires, they can be lifted 11 ft to give access to the upper seals.

The hoisting mechanisms are arranged to open or close either or both gates in 52 min.

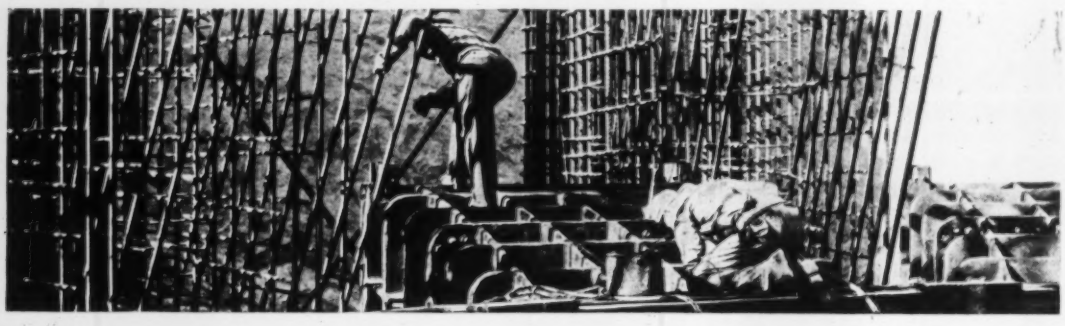
The combined weight of the four sets of gate hoists and stems is approximately 750,000 lb, while the stem guides weigh 226,000 lb more.

BULKHEAD GATES

For inspection and maintenance purposes the reservoir faces of the semi-steel liners of each of the 12 water passages leading to each upper and lower gate are provided with bronze seats to receive a cast-steel bulkhead gate to close each passage. Suitable bronze-faced guide grooves are provided in the buttress walls or fins of every tower which permit the transfer of the one complete set of 24 bulkhead gates to any tower, inserting them into their guide grooves and lowering them by special-cable operated dollies to their proper levels by means of a revolving electric crane installed in the top of each tower. The upper bulkhead gates weigh 9500 lb each, or 114,000 lb for the 12 required, while the lower bulkhead gates weigh 13,500 lb each, or 162,000 lb for the 12 required, a total of 276,000 lb for the upper and lower sets. The bronze bulkhead-gate guides inserted in the buttress walls were purchased on government plans and specifications No. 593-D and required 212,000 lb of bronze.

These bulkhead gates permit the unwatering of any tower, the inspection and maintenance of the cylinder gates, and their dismantling and removal for major repairs, if such procedure should ever be necessary. A heavy structural-steel maintenance platform has been provided which can be quickly transported to the tower being serviced and installed in the throat liner of any gate whenever need arises for major maintenance work.

In Part 2 of this paper, which will appear in a later issue, descriptions will be given of the 14-ft butterfly valves in the power house and the batteries of 72- and 84-in. needle valves with their 86- and 96-in. Paradox emergency gates.

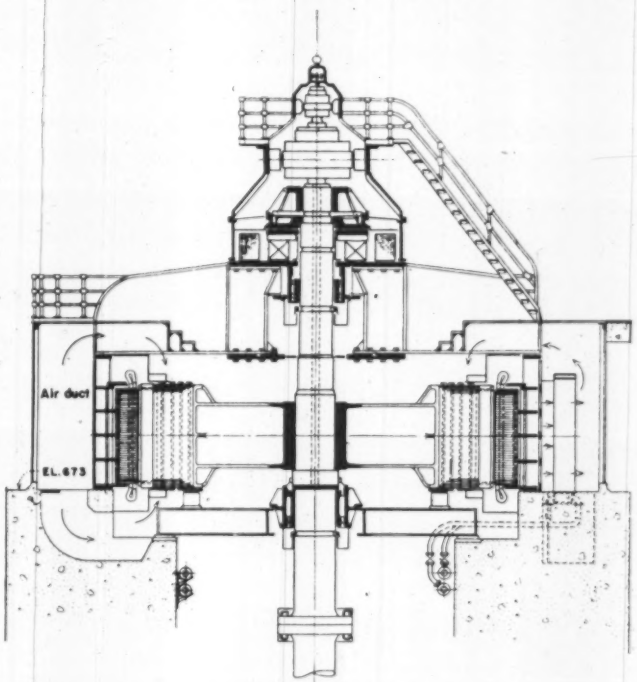


BOULDER DAM GENERATORS

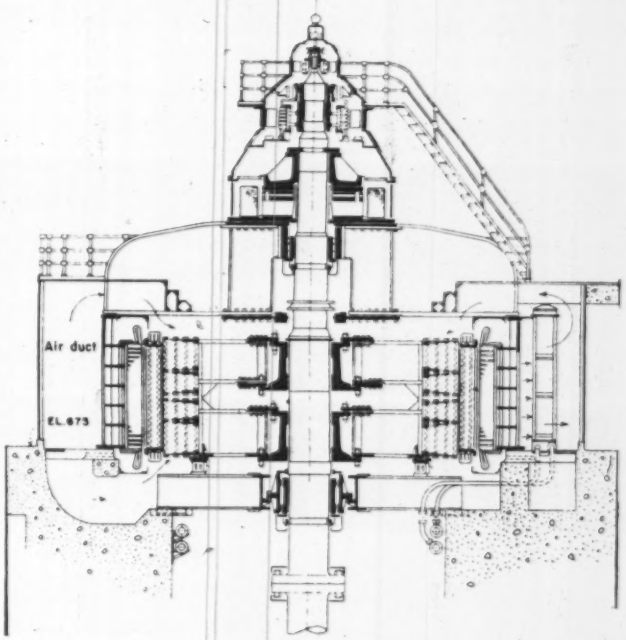
The Mechanical Features of the Largest Hydroelectric Power-Plant Generators in the World

By L. N. McCLELLAN[1]

THE BOULDER power plant is designed for an ultimate installation of 15 main generating units of 82,500 kva capacity each and two main generating units of 40,000 kva capacity each, making a total generating capacity of 1,317,500 kva. The initial installation, the machinery for which has been purchased and is now being manufactured, consists of four 82,500-kva units, two of which are being manufactured by the General Electric Co. and the other two by the Westinghouse Elec. & Mfg. Co.; and one 40,000-kva unit, which is being manufactured by the Allis-Chalmers Mfg. Co. The 82,500-kva generators are in several respects the largest hydroelectric generators in the world and while they involve no new features of design they are of interest because of their size and because of the special electrical and mechanical features incorporated in their design.



82,500-KVA WESTINGHOUSE GENERATOR, 150–180 RPM



82,500-KVA GENERAL ELECTRIC GENERATOR, 150–180 RPM

The power generated by the four 82,500-kva generators will be transmitted over two 287,500-volt transmission lines to Southern California, a distance of approximately 265 miles, and in order to satisfy the exacting service requirements and secure the greatest possible freedom from interruptions, the generators and transmission lines are being designed to afford a high degree of system stability. Special characteristics are incorporated in the design of the generators so that the transmission lines will have sufficient synchronizing power to hold the rotating apparatus at the two ends of the lines in step under extreme changes of load, whether caused by normal or abnormal system conditions. Each generator has a large flywheel effect, equivalent to 110,000,000 lb at a one-foot radius, which is about twice the flywheel effect which a generator of this rating of normal design would have. This is to prevent the speed of the unit from suddenly increasing when a short circuit occurs on a transmission line and thereby prevent the generator from getting out of synchronism with other rotating apparatus connected to the system. Low generator reactance tends to increase the system stability because it results in small instantaneous drop in voltage and change in phase of voltage when system short circuits occur. The normal transient reactance of generators of this capacity would be about 30 per cent, whereas the Boulder generators have a transient reactance of only 21 per cent. High short-circuit ratio also has a tendency to increase the system stability by reducing the demagnetization effect of short-circuit current, and the Boulder generators have a short-circuit ratio of 2.4, or about twice the short-circuit ratio which would be found in a generator of normal

[1] Chief Electrical Engineer, U. S. Bureau of Reclamation, Denver, Colo.

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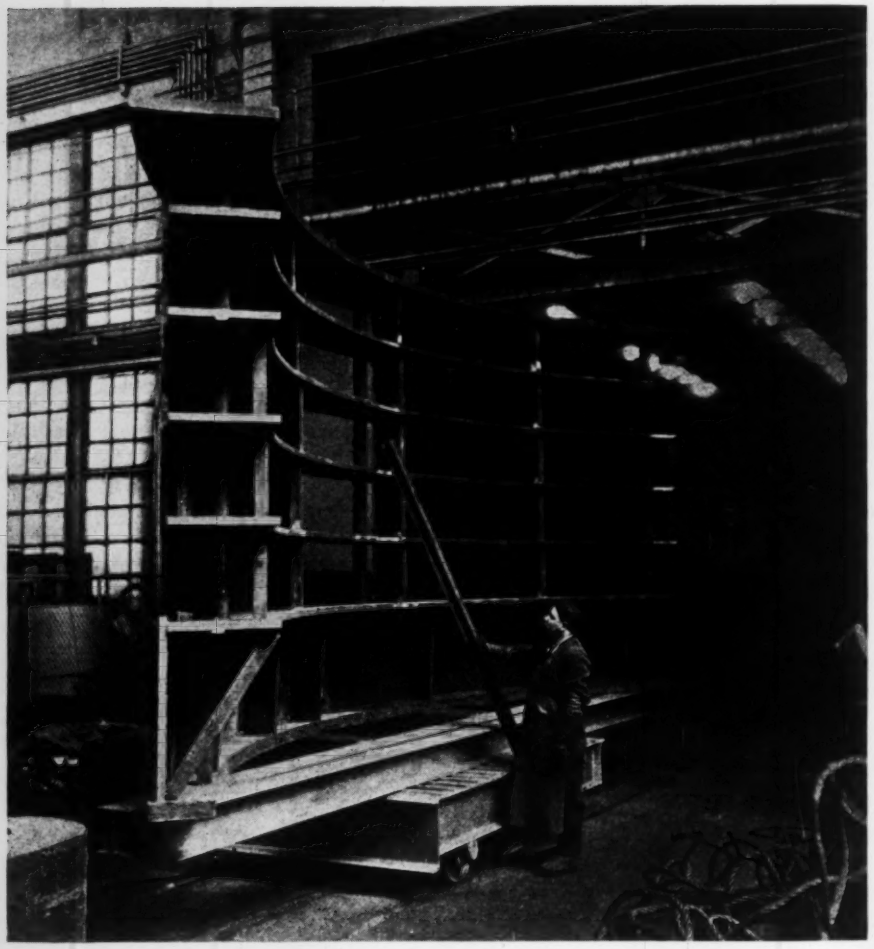


SECTION OF STATIONARY ARMATURE FRAME FOR 82,500-KVA GENERAL ELECTRIC WATERWHEEL GENERATOR

design. These special electrical and mechanical features of design all tend to increase the size of the generators and in physical dimensions they are as large as 125,000-kva machines of normal design. The generators are 40 ft in diameter, 32 ft in height above floor level, and each generator weighs approximately 2,000,000 lb.

They are designed to operate at either 50 cycles, 150 rpm, and 13,800 volts, or 60 cycles, 180 rpm, and 16,500 volts.

STATOR

The stator frames of the generators are entirely of fabricated construction and are made of rolled-steel plates cut by torch to the proper shape and dimensions and electrically welded together. Each stator frame is designed rigidly to support the armature core and windings and to carry the weight of the rotating parts of the generator and turbine and consists of a heavy vertical cylindrical steel wrapper plate the height of the frame, on the inside of which five equally spaced, horizontal, circumferential steel stiffener rings or ribs which support the stator core and windings are electrically welded. In addition to the five intermediate rings there are top and bottom plates of heavier cross-section which are also welded to the wrapper plate. The arms of the upper bearing bracket rest on and are rigidly bolted and doweled to the top plate, while the bottom plate rests on and is bolted and doweled to the foundation caps.

The armature core is built up of high-grade silicon-steel laminated punchings, insulated between layers by a coat of insulating enamel to reduce eddy-current losses, clamped tightly together by means of heavy through bolts with clamping flanges on each end. In the generators furnished by the Westinghouse Elec. & Mfg. Co. the laminations are assembled on heavy through bolts which, after being tightened so as to clamp the laminations firmly together, are electrically welded to the circumferential rings of the stator frame. In the generators furnished by the General Electric Co. the laminations are attached by means of dovetailed slots to vertical steel ribs which are electrically welded to the circumferential rings of the stator frame. The armature core is provided with air ducts through which air is forced by the fan action of the rotor. Each generator frame will be shipped in four sections to facilitate handling and the core and windings will be assembled at the destination.

The stator winding consists of machine-wound coils, accurately formed to size and shape. The conductors are of copper and are subdivided into a number of small strands of rectangular cross-section. Each strand is insulated from the adjacent strands in order to minimize the eddy-current losses. The coils are insulated with full Class B insulation in accordance with the standards of the American Institute of Electrical Engineers, consisting of mica tape impregnated with insulating compound and pressed while hot so as to obtain a compact and homogeneous insulation free from air-pockets. The coils are given a final protective covering of semi-conducting asbestos tape which serves to ground the outer surface of the coil to the stator core and thereby prevent corona discharge which otherwise might damage the coil insulation. The ends of the coils, where they project beyond the stator core, are supported mechanically by being securely lashed to insulated steel supporting

FABRICATED FRAME OF STATIONARY ARMATURE FOR 82,500-KVA GENERAL ELECTRIC WATERWHEEL GENERATOR

rings, and wooden spacer blocks lashed to the end turns to prevent deformation under short-circuit stresses. Embedded temperature detectors of the standard 10-ohm-resistance type are provided for the purpose of indicating the temperature of the windings.

UPPER AND LOWER BEARING BRACKETS

The upper bearing bracket is a heavy bridge member with a span of about 30 ft, which supports the upper guide bearing and the thrust bearing and carries the entire weight of the rotating parts of the generator and turbine, amounting to approximately 2,000,000 lb. It is designed to support this load with a very small deflection. The upper bearing bracket consists of eight radial arms of deep I-beam section, fabricated by fusion welding. The outer ends of the radial arms rest on and are bolted and doweled to the top circumferential ring of the generator frame, and the inner ends are rigidly attached by means of fitted bolts and keys to heavy horizontal circular steel plates forming the central hub of the bearing bracket.

The lower bearing bracket provides a rigid support for the lower guide bearing and also carries the air brakes and hydraulic jacks. It is entirely of fabricated construction, and is made of standard I-beam and channel sections with the necessary rolled-steel plate, all of which are electrically welded together. Both the upper and lower bearing brackets will be shipped disassembled to facilitate handling.

ROTOR

In order to provide the flywheel effect of 110,000,000 lb at one-foot radius, it was necessary for the rotor to have a large diameter and a deep rim. The rotors of the General Electric generators are 24 ft 8 in. in diameter, the rims are 8 ft 3 in. deep, and each rotor weighs approximately 1,165,000 lb. The rotors of the Westinghouse generators are 25 ft 3 in. in diameter, the rims are 7 ft 8 in. in depth, and the completely assembled rotor weighs approximately 1,080,000 lb. The rotor was the most difficult problem in connection with the mechanical design of the generators because of the tremendous centrifugal forces to be taken care of. When revolving at runaway speed, the stresses due to the centrifugal force are about 3.15 times the stresses at normal speed. The rotors are designed so that when operating at a maximum runaway speed of 320 rpm the stresses will not exceed two-thirds of the elastic limits of the materials used in their construction. The rim of the rotor is made of a large number of thin, high-strength, steel plates tightly clamped together by means of steel through bolts. The rim is of the free or floating type, that is, it is not restrained by the supporting spider which serves only to carry the weight of the rim of the rotor and of the field poles and to transmit the shaft torque to the rim. This construction relieves the center spider of all centrifugal stresses except those due to its own mass, and the stresses in the rotor can be definitely determined. The strength of the rim of the rotor would not be appreciably reduced by a flaw in any one of the large number of individual steel plates used in its assembly.

The spiders and rims of the rotors of the General Electric generators are built in two separate sections and are assembled on the shaft one above the other, while the rotors of the

SHAFT FOR 82,500-KVA WATERWHEEL GENERATOR

Westinghouse generators, which are not as deep as those of the General Electric machines, are assembled in a single section. The center spiders differ considerably in the two designs; the spider of the Westinghouse generator is a one-piece steel casting with eight radial arms of I-beam section cast integrally with a central hub which is shrunk on and keyed to the shaft. The center spider of the General Electric generator is a fabricated structure built in two sections, each of which consists of ten fabricated radial arms of heavy I-beam section 3 ft deep, welded to a solid steel plate the diameter of which is the same as the inside diameter of the rim of the rotor. The radial arms are attached by means of fitted bolts to two cast-steel hubs which are shrunk on and keyed to the shaft. The lower spider carries the braking plates and supports the weight of the lower half of the rotor rim while the upper spider carries the weight of the upper half of the rotor rim. In both designs the torque is transmitted from the shaft to the rotor rim by keys between the spider hub or hubs and the shaft, and by keys between the ends of the radial arms of the spider and the laminated rim. These latter keys are radial and therefore do not restrain the rotor rim from expanding under the stress due to centrifugal force. The laminated rim of the rotor and the field poles will be assembled at destination in order to facilitate handling and loading.

The pole pieces are attached to the rim of the rotor by means of dovetails fitting into corresponding slots in the rotor rim and the poles are secured by driving taper keys between the dovetails of the poles and the corresponding slots in the rim of the rotor. The keys are cut off flush with the rotor rim after they have been driven to proper tightness and are then covered by plates bolted to the rotor rim which prevent the keys as well as the poles from becoming displaced. The pole pieces are built up of thin laminations tightly clamped together by bolts passing through the laminations and supporting end plates. The field coils are made of heavy copper strap wound on edge with the turns insulated from each other by asbestos insulation. Small rugged coil springs placed in recesses in the rim of the rotor bear against the steel collars of the inner ends of the field coils and maintain a pressure on the field coils to keep them tight on the poles and compensate for any shrinkage of the insulation. The field poles are provided with an amortisseur winding of the non-continuous low-resistance type.

WELDING

All welding in connection with the fabrication of the generator frames, bearing brackets, and rotors is of the electric-fusion type, and automatic or machine welding is required for all circumferential welds in so far as practicable. Welds which are subjected to tension or bending are required to be made by the shielded-arc method, using heavily coated welding electrodes or a suitable flux, or gaseous envelope to shield the arc and molten metal and prevent oxidation. The welders are required to be capable of passing the qualification tests for the type of welding they perform; all bare-electrode welders are required to pass the qualification test outlined in the report of the structural steel welding committee of the American Bureau of Welding and all heavily covered-electrode welders are required to pass the qualification test outlined in Par. U-69 of the A.S.M.E. Rules for Unfired Pressure Vessels.

THE SHAFT

The main shafts are made of forged heat-treated carbon steel, with coupling flanges forged integrally with the shaft. The shaft material has an ultimate strength of 75,000 lb per sq in. and a yield point of from 35,000 to 37,000 lb per sq in., and under normal operating conditions the material is stressed to not more than 4500 lb per sq in. The main generator shafts are 38 in. in diameter and weigh more than 100,000 lb. The shafts of the General Electric generators are 36 ft 3 in. in length and the shafts of the Westinghouse generators are 32 ft 8 in. in length. Each shaft has an inspection hole approximately 6 in. in diameter bored axially throughout its length. The weight of the rotating parts is carried by a cast-steel thrust collar attached to the upper end of the shaft by means of a circular key.

BRAKES AND LIFTING JACKS

The generators are provided with air-operated brakes which have sufficient capacity to bring the rotating parts of the unit to a stop from a speed of 75 rpm within 7.5 min after the brakes are applied without field excitation of the generator and with leakage through the turbine gates sufficient to produce one per cent of full rated generator torque. The brakes consist of a number of individual cylinders and brake shoes spaced around and beneath the lower face of the rim of the rotor. They are

designed to be used as hydraulic jacks for lifting the rotating parts of the unit. The brakes operate with an air pressure of 60 to 75 lb per sq in. and when used as hydraulic jacks for lifting the rotating parts off the thrust bearings oil at a pressure of 2000 lb per sq in. is used in the cylinders. Hinged-metal blocks are provided for insertion between the brake ring and the arms of the lower bearing bracket to support the generator rotor in the lifted position after it has been raised by means of the hydraulic jacks.

THRUST BEARING

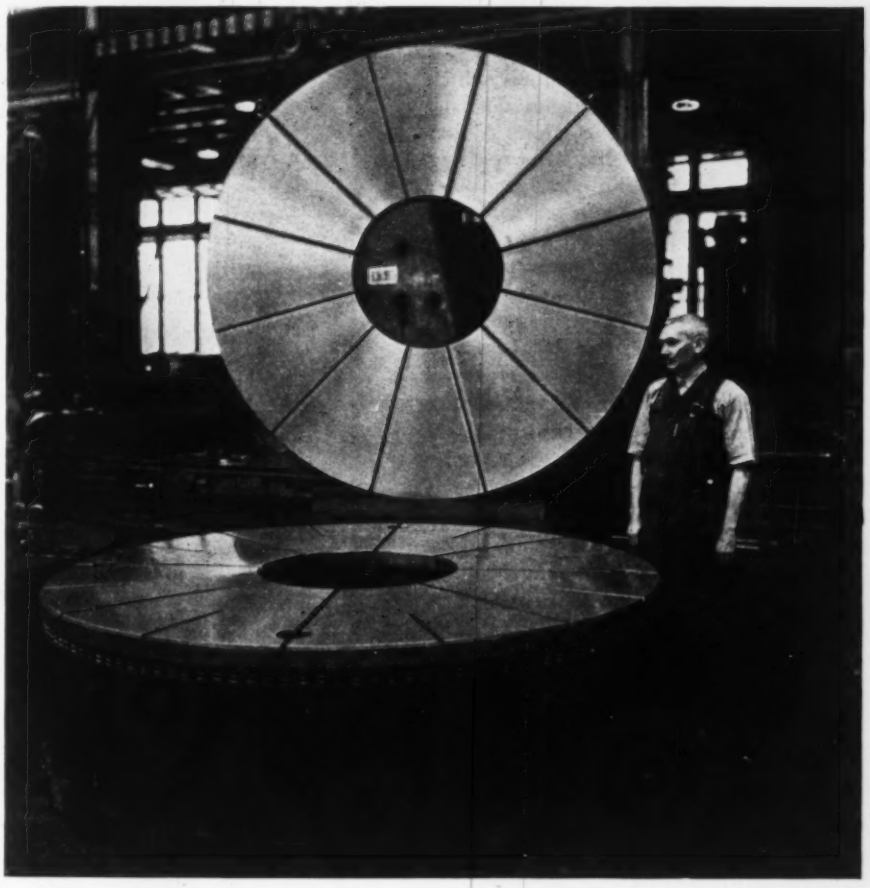
The combined weight of the rotating parts of the generator and turbine including the unbalanced hydraulic thrust of the turbine runner is carried by a thrust bearing supported by the upper bearing bracket. The thrust bearings of the two Westinghouse generators are of the Kingsbury type, consisting of a rotating plate attached to a thrust collar keyed to the upper end of the main shaft and eight stationary babbitt-surfaced bearing shoes, each supported by a jack screw by means of which each shoe may be adjusted independently so as to equalize the load on all shoes. The Kingsbury-type thrust bearings are 84 in. in diameter and have a rated capacity of 1,700,000 lb. The two General Electric generators have thrust bearings of the spring type, which consist of a rotating steel plate, rigidly supported by a thrust collar attached to the upper end of the main shaft, and a relatively thin stationary steel plate with a babbitted bearing surface supported by a large number of pre-compressed coil springs. The stationary plate has a radial slot to increase its flexibility and is sufficiently flexible to yield slightly under high local pressures so as to distribute the load evenly over the entire bearing surface. The spring-type thrust bearings are 90 in. in diameter and have a rated capacity of 1,854,800 lb. The thrust bearings are provided with temperature detectors imbedded in the bearing metal for indicating the temperature of the bearings. A thermometer is located in the hottest oil leaving the bearings. The unit is arranged to be shut down automatically if the temperature of the oil exceeds a predetermined amount.

GUIDE BEARINGS

The generators are of the conventional two-guide-bearing type with one guide bearing located above and the other below the rotor. The guide bearings are of the babbitt-lined sleeve type, designed so that the bearing liners may be removed without disturbing the rest of the generator. Each guide bearing is provided with a temperature detector imbedded in the bearing metal for the purpose of indicating the temperature of the bearing and automatically shutting the unit down if the temperature exceeds a predetermined amount. The upper guide bearings of both designs are 30 in. in diameter and 20 in. in length and the lower guide bearings are 38 in. in diameter and 32 in. in length.

LUBRICATION

A unit system is provided for the lubrication of each generator. Oil from the guide bearings and from the thrust bearing flows by gravity to an oil reservoir. Duplicate lubricating-oil circulating pumps are provided for each unit, one driven by an alternating-current motor receiving its power supply from the station service and the other driven by a direct-current



92-IN. SPRING THRUST BEARING OF 77,500-KVA GENERAL ELECTRIC WATERWHEEL GENERATOR

(Thrust bearings of the 82,500-kva General Electric Generator for Boulder dam are of this design but are 90 in. in diameter.)

motor receiving its power supply from the station storage battery. These oil pumps circulate oil from the reservoir to the two guide bearings and also to the thrust bearing, if required, although normally oil is not circulated through the thrust bearing continuously. The alternating-current pump is normally in service, and the direct-current pump is a standby unit arranged to start up automatically in case of a failure of the oil supply from the other pump.

The thrust bearing is submerged in a bath of lubricating oil and circulation is effected by the centrifugal pumping action of the bearing itself. Cooling coils through which cold water is circulated remove the heat from the oil in the thrust-bearing housing. After passing over the cooling coils the oil flows to the center of the bearing and from there passes radially outward through oil grooves in the stationary and rotating plates. Sight-flow gages and thermometers are provided in the oil lines, both on the main generator floor and in the governor gallery. Automatic interlocks are arranged to prevent the unit from starting until circulation of lubricating oil is normal and to shut the unit down in case the supply of lubricating oil fails.

VENTILATING SYSTEM

Each generator is provided with an enclosed ventilating system complete with air ducts, surface coolers, and housing. The enclosed system of cooling was adopted because it affords complete control of the temperature of the generator; it prevents dirt and foreign material from coming in contact with the windings, thereby prolonging the useful life of the generator and reducing the cost of maintenance; and it permits the use of carbon-dioxide-gas fire-extinguishing equipment to protect the generator from internal fires. Circulation of air is accomplished by means of fans attached to the generator rotor, which force the air through the openings in the stator core and thence to the surface coolers where the air is cooled. After passing through the coolers the air divides in the outer housing, part of it passing up over the top of the stator frame between the arms of the upper bracket and down into the rotor, while the remainder passes down and then inward through ducts in the concrete foundation immediately below the base of the stator frame and thence up to the lower part of the rotor. Each generator has eight surface air coolers arranged symmetrically around the periphery of the machine. Each cooler consists of a number of Admiralty-metal tubes through which cold water is circulated and over which the circulating air passes. The tubes are connected to water boxes at both ends which have removable cover plates so that the tubes may be cleaned and inspected without disturbing the pipe connections. The coolers are designed so that any tube may be removed and replaced without disturbing the other tubes. The surface coolers are designed for a maximum working pressure of 50 lb per sq in. and they are tested at the factory with a hydrostatic pressure of 75 lb per sq in. The coolers have sufficient capacity to cool the generator with only seven coolers in service. One cooler can be taken out of service at any time for maintenance or repairs without interfering with the output of the generator.

STATIONARY-ARMATURE COIL FOR 82,500-KVA GENERAL ELECTRIC WATERWHEEL GENERATOR

FIRE-EXTINGUISHING SYSTEM

The generators are provided with carbon-dioxide-gas fire-extinguishing equipment with automatic control designed to maintain a concentration of not less than 25 per cent of carbon-dioxide gas in the generator housing for a period of 45 min. If an internal fault develops in the generator winding, causing the differential relays to operate, or if the thermal relays installed in the hot air ducts of the generator ventilating system function due to excess temperature, the generating housing will be filled automatically with carbon-dioxide gas to the necessary degree of concentration. Provision is also made for the manual admission of carbon-dioxide gas to the generator housing by means of control stations of the "break-glass" type, located at the switchboard and on the generating housing.

EXCITERS

Each generator has an individual direct-connected 250-volt main exciter mounted on top of the thrust-bearing housing, and a direct-connected 250-volt pilot exciter mounted on top of the main-exciter frame. The main exciters of the two generators furnished by the General Electric Co. have a rated capacity of 564 kw, and the pilot exciters have a rated capacity of 18 kw. The two generators furnished by the Westinghouse Elec. & Mfg. Co. have main exciters rated at 425 kw and pilot exciters rated at 7 kw.

A small direct-connected alternating-current generator of the permanent-magnet type mounted on top of the pilot exciter supplies power for the operation of a motor which drives the speed-responsive element of the governor.

Each generator has an individual automatic voltage regulator of the non-continuously vibrating, high-speed type, designed to respond to changes in voltage in a period of three cycles. The field of the main generators, as well as the exciters and voltage regulators, are all designed for high-speed response excitation, which is an important factor in increasing the stability of the power system.

ACKNOWLEDGMENTS

The 82,500-kva generators for the Boulder power plant were designed by the General Electric Co. and the Westinghouse Elec. & Mfg. Co. in accordance with specifications prepared by the U. S. Bureau of Reclamation. The author is indebted to the engineers of both these companies for valuable suggestions and help in connection with the preparation of this paper. All designs prepared by the Bureau are under the general direction of J. L. Savage, chief designing engineer, all engineering and construction work is under the direction of R. F. Walter, chief engineer, with headquarters at Denver, Colo., and all activities of the bureau are under the general charge of Dr. Elwood Mead, Commissioner of Reclamation, with headquarters at Washington, D. C.

TURBINES *for* BOULDER DAM

Mechanical Features of the Largest Hydraulic Turbines in the World

By IREAL A. WINTER[1]

THE BOULDER power plant is located at the Boulder dam site on the Colorado River, approximately 25 miles southeast of Las Vegas, Nevada. The power plant is immediately downstream from the dam and forms a U-shaped structure with the base of the "U" across the river on the downstream toe of the dam and one wing on each side of the river.

The main generating units will be installed in the wings of the power plant and the central or connection portion will contain station service units, sump pumps, machine shop, and control and auxiliary equipment. The power plant is designed for the ultimate installation of fifteen 115,000-hp turbines driving 50-cycle main generators of 82,500 kva capacity each, and two 55,000-hp turbines driving 60-cycle main generators of 40,000 kva capacity each. The initial installation comprises four 115,000-hp units and one 55,000-hp unit. The 82,500-kva units are designed for possible future operation at 60 cycles by changing the turbine runners.[2]

The general arrangement of the power plant, dam, intake towers, headers, penstocks, and outlets is shown in Fig. 1.

There are four intake towers, two on each side of the river immediately upstream from the dam. Each tower controls one-quarter of the supply of water for the turbines in the power plant and each tower has two cylinder gates,[3] by means of which the water can be shut off from any one of the four main penstock headers without interfering with the operation of the other three headers. The lower cylinder gates are at elevation 895 and the upper cylinder gates are at elevation 1045. The intake towers are provided with means for placing emergency bulkhead gates over the entrance to the cylinder gates so that the latter may be unwatered for inspection and maintenance purposes. Trashracks are provided over the entrances to the intake towers.

Steel penstock headers 30 ft in diameter installed in two of the 50-ft diameter diversion tunnels and in the two main penstock header tunnels will conduct the water from the intake towers to the individual penstocks and to outlet valves.[4] Each 30-ft main penstock header connects with four branch penstocks, each 13 ft in diameter, which will conduct water to the main turbines. The two 55,000-hp turbines, however, are connected to the downstream 13-ft branch penstocks from

[1] Engineer, Bureau of Reclamation, Denver, Colo.

[2] See paper by McClellan in this same issue, pp. 409–414.

For presentation on "Reclamation Bureau Night" at the Semi-Annual Meeting, Denver, Colo., June 25 to 28, 1934, of The American Society of Mechanical Engineers. Published by permission of the Bureau of Reclamation.

[3] See paper by Kinzie in this same issue, pp. 387–408.

[4] The design and fabrication of these steel pipes are described in a paper by C. M. Day and Peter Bier to be presented at the Semi-Annual meeting, Denver, Colo., June 25 to 28, 1934, of The American Society of Mechanical Engineers.

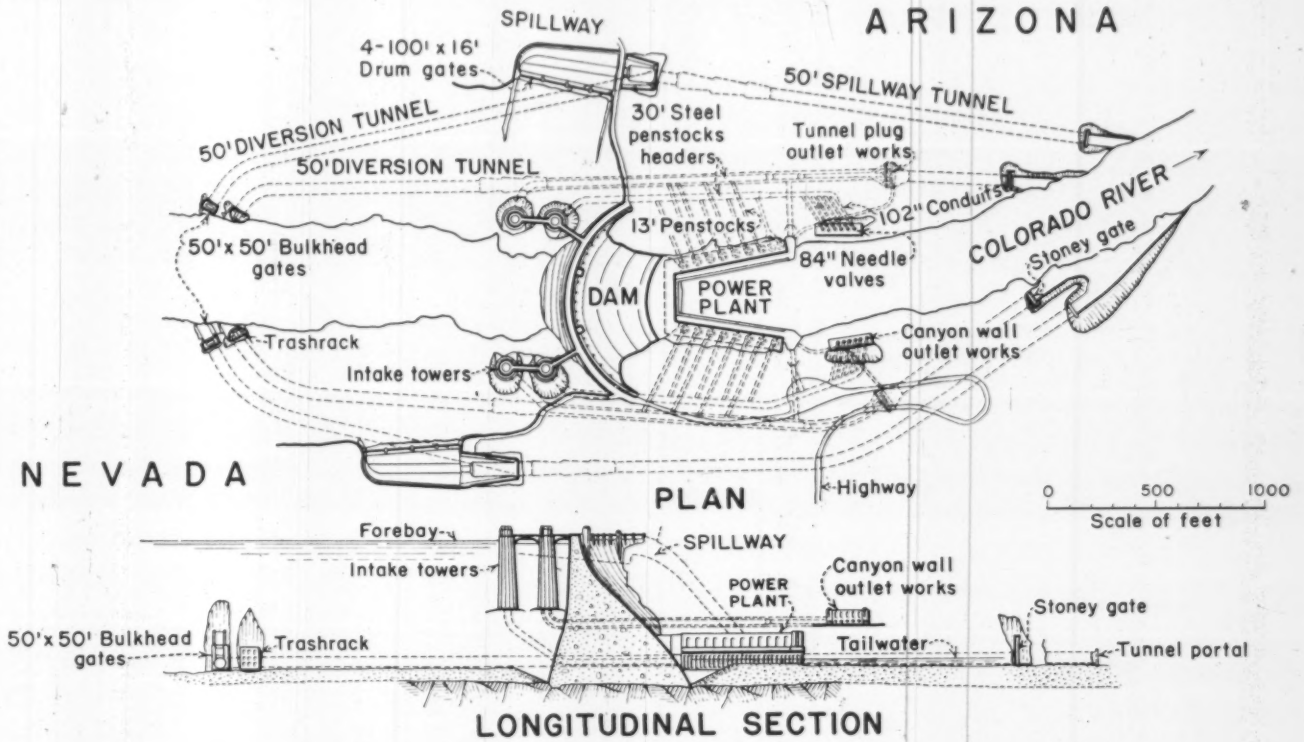


FIG. 1 GENERAL LAYOUT OF BOULDER DAM AND POWER PLANT

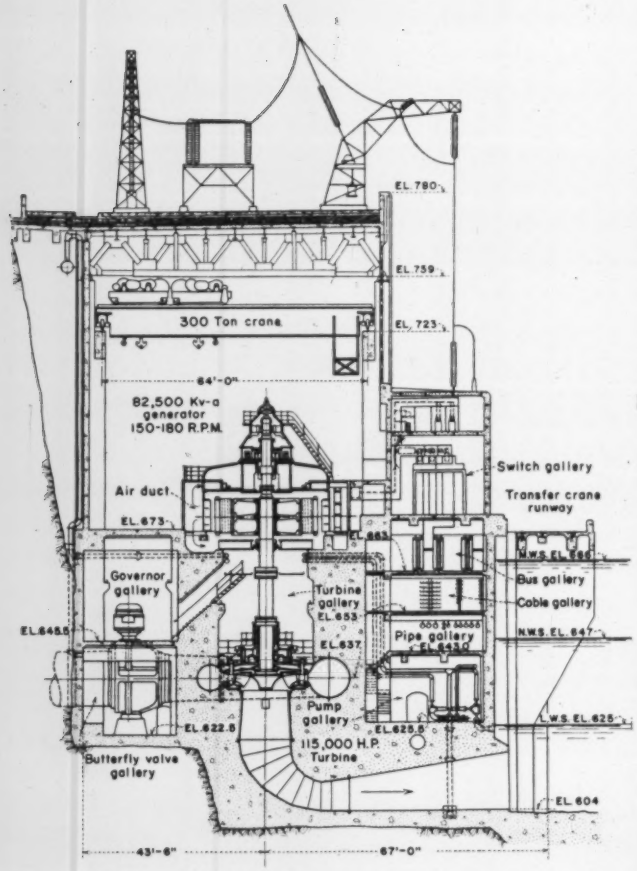


FIG. 2 TYPICAL CROSS-SECTION THROUGH THE BOULDER POWER PLANT

the upper or penstock header tunnel on the Arizona side. The four initial 115,000-hp turbines are connected to branch penstocks from the upper and lower tunnels on the Nevada side of the river. A butterfly-type shut-off valve is installed at the inlets of the turbine casings.[5]

HYDRAULIC CONDITIONS

The Boulder Canyon project is designed and will be operated to augment the water supply of the southwestern states for domestic, industrial, and irrigation uses, and to provide flood control, with the incidental generation of power. Boulder dam will impound water from a drainage area of 170,000 sq miles. Of the total capacity of 30,500,000 acre-ft, 4,000,000 acre-ft lies below level of the entrance to the power penstocks.

The maximum flow at the Boulder dam site is 250,000 cfs and the minimum flow 2300 cfs. It is contemplated that when the Boulder Canyon project is completed the maximum flood to be passed will be 75,000 cfs, with a regulated minimum flow of about 8000 cfs.

The elevation of the water surface in the reservoir will fluctuate from a maximum at 1229, with flood storage capacity completely full, to a minimum at about 1000. Under present conditions of river bed, the surface of the water in the river at the dam site, with a low-water flow of 12,000 cfs, is at about elevation 647, and with a discharge of 100,000 cfs, the water surface in the river immediately below the dam is at about elevation 666.

It is expected that the loose material in the river bed below the dam will be gradually eroded by the clear water released from the reservoir and that this will result in the future in a somewhat lower elevation of tailwater than exists at the present time. In order to provide for this possibility, the power plant is designed for a minimum tailwater surface at elevation 625.

The center lines of the turbine distributors are at elevation 637 and the elevation of the water surface in the tailrace will vary from about 28 ft above to about 12 ft below the center line of the turbines. (See Fig. 2.) The average elevation of the water surface in the tailrace with channel conditions as they are at present will be from 10 to 12 ft above the center line of the turbine distributors.

The net effective head under which the turbines will operate may vary from a minimum of about 360 ft to a maximum of 590 ft. These extreme head conditions will occur only at infrequent intervals and will be of short duration, and for 90 per cent of the time the net effective head will be between 450 and 560 ft, and the average net effective head will be about 530 ft.

HYDRAULIC TURBINES

The large storage capacity will enable most of the river flow at the Boulder dam site to be converted into prime power. The regulated flow will continuously generate 700,000-hp with the turbines loaded at the point of best efficiency and at an operating head of 530 ft. To meet load conditions for this

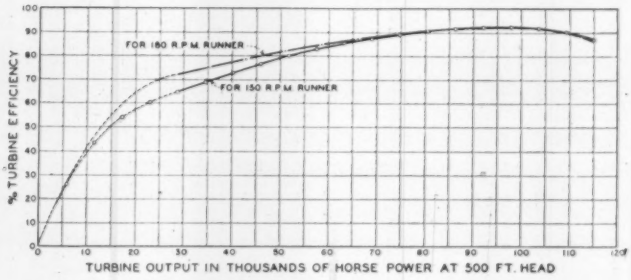


FIG. 3 CURVES SHOWING EXPECTED PERFORMANCE OF 115,000-HP TURBINE AT THE RATED HEAD

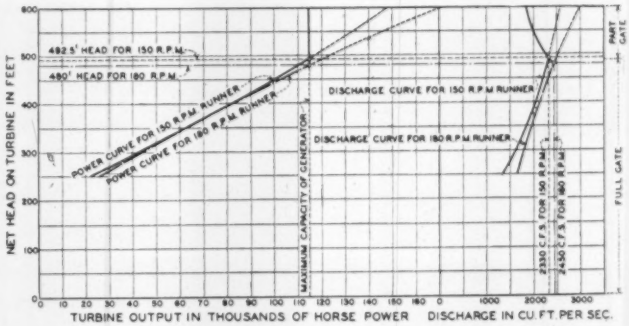

NET HEAD ON TURBINE IN FEET
482.5' HEAD FOR 150 R.P.M.
480' HEAD FOR 180 R.P.M.
DISCHARGE CURVE FOR 150 R.P.M. RUNNER
DISCHARGE CURVE FOR 180 R.P.M. RUNNER
POWER CURVE FOR 150 R.P.M. RUNNER
POWER CURVE FOR 180 R.P.M. RUNNER
MAXIMUM CAPACITY OF GENERATOR
PART GATE
FULL GATE
2330 C.F.S. FOR 150 R.P.M.
2450 C.F.S. FOR 180 R.P.M.
TURBINE OUTPUT IN THOUSANDS OF HORSE POWER
DISCHARGE IN CU. FT. PER SEC.

FIG. 4 EXPECTED HORSEPOWER OUTPUT AND TURBINE DISCHARGE IN CUBIC FEET PER SECOND FOR 115,000-HP TURBINE

power, sufficient units will be installed to bring the turbine installation up to 1,800,000 hp.

TYPE

The turbines selected for the Boulder power plant are of the vertical-shaft single-runner Francis type with spiral casing. The four initial 115,000-hp turbines will operate at a speed of

[5] The design and fabrication of these valves are described in a paper by Phillip A. Kinzie to be presented at the Semi-Annual Meeting, Denver, Colo., June 25 to 28, 1934, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

150 rpm but are so designed that they may be operated safely and satisfactorily at a speed of 180 rpm, with no major change in any part of the apparatus except the installation of new runners suited to the higher speed. (See Figs. 3 and 4.) The turbines are designed for the two speeds to permit the generation of 50- and 60-cycle alternating current without alteration in the generators.

The physical dimensions of the hydraulic passages for the 115,000-hp turbines are of normal size; the scroll-case entrance diameter being 10 ft, the discharge diameter of the runner approximately 13½ ft, the height of the distributor case 19 in., and the shaft diameter 38 in. In mechanical structure and horsepower capacity, however, these turbines distinctly set a precedent for hydraulic prime movers.

HYDRAULIC DESIGN

The turbines as designed will have a scroll-case inlet velocity of 16 per cent of the spouting velocity under the designed head. (See Fig. 5.) The cross-sectional area of the scroll case progressively increases around the turbine relative to the increment flow into the speed ring, producing in effect a negative acceleration. The guide case will consist of 24 fixed stay vanes in the speed ring, with 24 movable wicket-type gates for regulating the power output of the turbine. The velocity of the water at the tip of the wicket gates when the unit is operating at the point of best efficiency will be approximately 50 per cent of the spouting velocity under the designed head.

The value of ϕ for the entrance edge of the runner blade at the center line of the distributor will be 67½ for the 150-rpm runner and 81 for the 180-rpm runner. There will be 21 buckets to each runner with area and directional changes to produce a final discharge velocity at the throat of the draft tube of 25 fps. The energy in velocity head of the water delivered to the throat of the draft tube will be approximately 2 per cent of the total energy available.

The turbine discharge to the tailrace will be through a flattened-type of elbow draft tube. The rate of negative acceleration of the draft-tube expansion in the elbow section (see Fig. 5) will be 5 ft per sec per sec. The residual velocity at the draft tube orifice will be 5 ft per sec when the turbine is

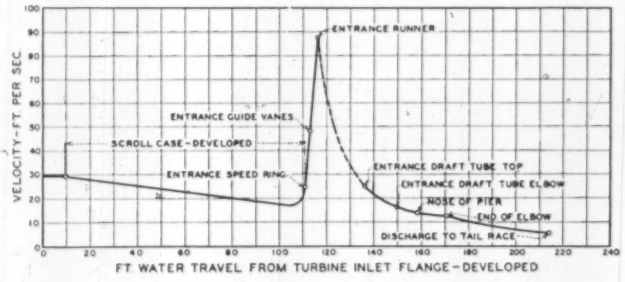


FIG. 5 CURVE SHOWING VELOCITY OF WATER THROUGH 115,000-HP TURBINE

operating at the point of maximum efficiency and at the rated head of 492½ ft. The residual velocity represents a final rejection of one-tenth of one per cent of the total available energy.

Water leakage around the shroud of the runner and between the crown plate of the runner and the turbine cover plate will be reduced to a minimum by the use of two seal rings of the non-labyrinth type. (See Fig. 6.) When the unit is operating as a synchronous condenser the runner seal chamber will be supplied with penstock water through two 4-in. pipes capable of delivering about 3 cfs to each seal chamber. The sealing water under these conditions is essentially a cooling medium to pre-

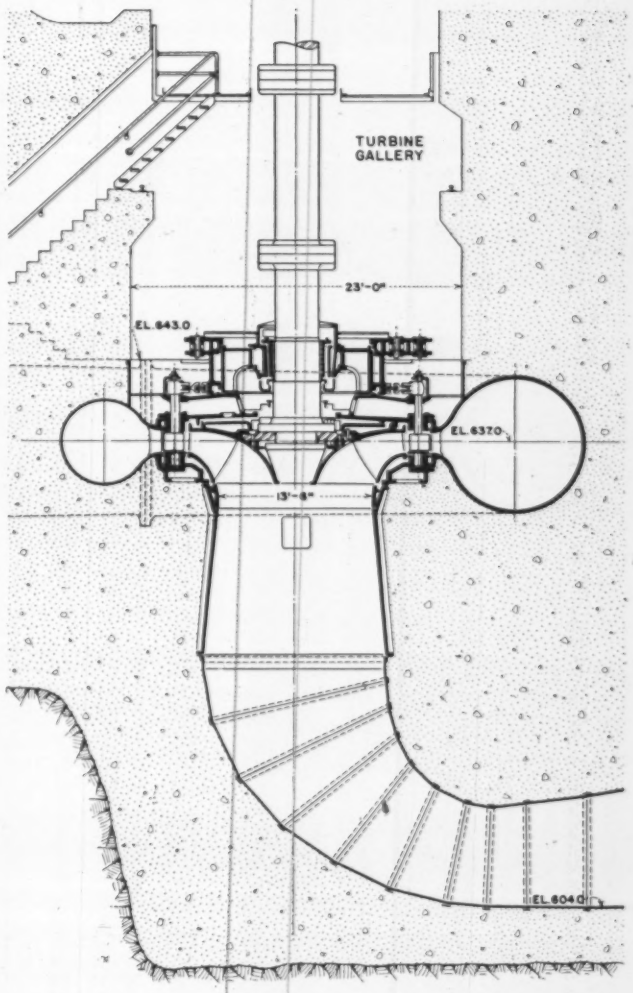


FIG. 6 CROSS-SECTION THROUGH 115,000-HP TURBINE

vent heating and seizing of the runner bands in case the stationary and revolving metals come in contact.

All projections and pockets in the runner seal chambers have been eliminated to reduce hydraulic losses to a minimum. The chamber between the crown of the runner and the bottom of the turbine cover plate is provided with a smooth plate deflector which will reduce materially the amount of water circulated by the runner crown and will also reduce the frictional load. This deflector apron forms a stationary channel for the leakage-water discharge near the center of the runner.

The electrical transmission requirements at the Boulder power plant are such that it may be desirable to operate the units as synchronous condensers. Special consideration had to be given to the problem of evacuating the draft tube when the unit is operating under these conditions, as the turbine runners are located below normal tailwater. This feature is considered essential as it is estimated that approximately 3000 kw will be required to drive the turbine runner in dead water, as compared to approximately 500 kw with the runner case empty. It was considered highly undesirable to drive the unit at full-speed-no-load with water through the turbine gates because of the great waste of water and the possibility of increased pitting of the metal parts of the turbine under these conditions.

For motoring purposes the water in the draft tube will be depressed to a point 5 ft below the bottom of the runner.

The penstock butterfly valve will be closed and the turbine scroll case drained down to the bottom of the distributor plate. The depressing of the water level in the draft tube will be effected by means of compressed air supplied from the house-service air-compressing plant. Two compressors each delivering 350 cu ft of free air per minute at 100 lb gage pressure will be required to evacuate the unit in $7^1/_2$ min under maximum high tailwater conditions.

The elevation of the water surface in the draft tube will be controlled by means of a float-operated air valve with water-level connection to the roof of the draft tube in the horizontal

FIG. 7 SPIRAL-CASING CASTING FOR 115,000-HP TURBINE

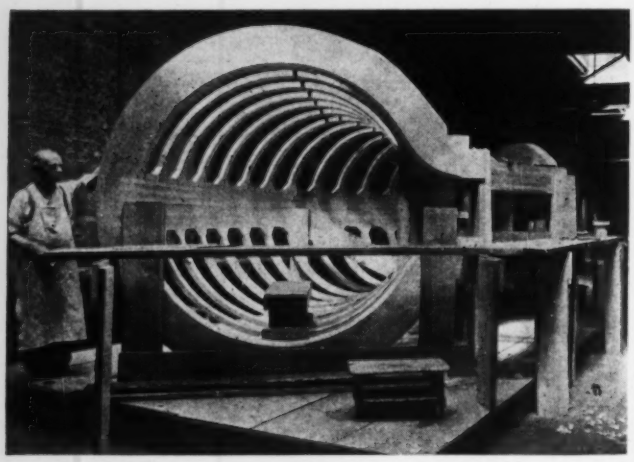


FIG. 8 SCROLL-CASE PATTERN FOR 115,000-HP TURBINE

discharge section. Pressure in the runner case and the top of draft tube will be transmitted to the automatic valve through a 3-in. pipe. The automatic float valve under these conditions will admit sufficient air to the turbine cover plate to maintain the water level at a predetermined point.

Provisions have been made for the admission of air through the cover plate at atmospheric pressure should the river bed be eroded below the power house. This air valve is of a conventional type, automatically operated from the turbine shifting ring, and will be provided with a pipe connection leading to the outside of the power house to remove from the turbine pit the objectionable noise of the air intake.

The turbine casing will be filled by means of two internal differential needle valves, 12 in. in diameter, located on the penstock butterfly valve. The entrained air in the turbine scroll case will be discharged through two 4-in. atmospheric relief valves designed to close automatically when the air has been completely expelled from the highest point in the conduit.

MECHANICAL DESIGN

The neat lines of the water passages for the Boulder turbines are of normal dimensions. The precedent established in connection with the mechanical design of these machines is due to a combination of the linear dimensions of a normal turbine and an internal pressure of 300 lb per sq in. It would have been impracticable to build these machines of larger capacity without the use of special steel and alloys to hold the metal thicknesses to reasonable dimensions.

All essential parts of the turbines are made of cast steel with allowable design stresses not to exceed 7500 lb per sq in. under normal operating conditions. For emergency conditions, stresses up to one-third of the yield point of the metal were allowed under the specifications. These limitations were considered not too conservative in view of the importance of the machines.

A survey of current methods of flange design showed that a method was lacking for such conditions as will exist in the turbine-scroll-case and relief-valve flanges.

One of the more common methods of flange design is based on the joining of a cylinder under hydrostatic pressure to a ring. The ring is considered to be heavy enough so that its only action is one of dishing or tilting, with the faces remaining straight as shown in Fig. 9. The outer ends of the flanges are not in contact. These conditions may exist where a metal ring-type gasket is used and where the bolts are not highly prestressed. The effect of a taper between the flange and the cylinder is not considered. This method may be referred to as the ordinary method.

The flanges for the turbine and relief valve are provided with round rubber gaskets placed in grooves machined in the flanges, and to obtain a water-tight fit the flanges will be drawn tightly together by prestressing the bolts to a stress above that which would exist due to the working loads. Under these conditions it was assumed that the flanges were brought into contact at the center line of bolts, and a theory was developed which was based on the joining together of a cylinder, a tapered section, and a circular plate fixed at the center line of bolts, as shown in Fig. 9. This will be referred to as the more accurate method.

As an analysis by the more accurate theory is laborious, an attempt is being made to develop an approximate method as shown in Fig. 9. The effect of internal pressure is neglected, the longitudinal pull only being considered, and an effective thickness of pipe is used to approximate the effect of a taper. This method is comparatively simple to use, and results obtained by it have been compared favorably with results obtained by the more accurate method. Sufficient cases have not been worked out by both methods, however, to warrant its use in all cases.

When a flange is analyzed by the ordinary and more accurate methods, a comparison of results shows that by using the ordinary method higher stresses at the junction of the flange and pipe and lower stresses in the flanges are obtained.

RUNNER

The turbine runner will be made of cast steel. It is designed and constructed to withstand safely the stresses due to operation at runaway speed under conditions of maximum head, including pressure rise with turbine gates wide open and with no load on the generator. The runner will have sufficient strength to support its own weight and the weight of the turbine shaft when the latter is disconnected from the generator shaft and the runner is resting on a ledge or shoulder in the

draft tube. The runner will be attached to the turbine shaft with a flanged connection using body-bound bolts. The finished runner with wearing rings attached will be statically balanced in the contractor's shop.

SHAFT

The turbine shaft is made of forged open-hearth carbon steel, heat treated, with an ultimate strength and yield point of 75,000 and 37,500 lb per sq in., respectively, an elongation in 2 in. of 22 per cent, and a reduction of area of 35 per cent. The shaft is 38 in. in diameter and is designed to operate at any speed up to the full runaway speed of 350 rpm without vibration or objectionable distortion.

A graphical diagram used in checking the shaft deflection and critical speed under normal and emergency conditions is shown in Fig. 10. In determining the first, or lowest, critical vibrating speed of the shaft, the varying diameters and loadings on the shaft made an analytical solution very long and tedious. As a result, the graphical solution shown in Fig. 10 was used. This method is based on the theory that a very close approximation of the first critical speed is given by the equation

$$N_c = \frac{\sqrt{g}}{2\pi}\sqrt{\frac{\Sigma(F \times \Delta_F)}{\Sigma(F \times \Delta_F^2)}}$$

where the F's are the gravity forces of the various masses, and the Δ_F's are the corresponding static shaft deflections that they cause when taken as acting upon the shaft in opposite directions in succeeding spans between bearings. The center bearing reaction under these conditions was determined by Maxwell's theorem of reciprocal deflections, and by the use of vector and funicular polygons to integrate graphically the differential equations of bending moments and deflections. The same method of graphical integration was then applied to determine the final deflection curve from which the critical speed was calculated.

The turbine shaft is made in two sections to facilitate handling the various turbine parts without dismantling the generator and the two sections will be connected by means of a male and female flanged bolted coupling. A portable power-house crane is provided in the turbine gallery for handling the hydraulic machinery. All coupling bolts will be body bound, forced fit, and of dimensions consistent with the remainder of the design.

GUIDE BEARINGS

The main turbine guide bearing is of the babbitt oil-lubricated type, located above the runner. The bearing support is anchored to the turbine cover plates and the bearing load is transmitted to the turbine speed ring and scroll case. The guide bearing is provided with three thermostat relays, one to each segment, arranged to close contact upon a predetermined rise in the temperature of the bearings. Leakage water from the shaft stuffing box will be prevented from entering the lubricating-oil system by means of a shedder ring placed around the shaft.

Two independent motor-driven positive-displacement rotary-type oil pumps are provided for circulating the lubricating oil. One of the oil pumps will be driven by a 60-cycle, 440-volt alternating-current motor and the other pump will be driven by a 250-volt direct-current motor which will be supplied from the station storage battery. Normally the oil will be circulated by the alternating-current-motor pumping unit, but on failure of the oil pressure or failure of the alternating-current supply the direct-current motor is arranged to start automatically and supply oil to the bearings.

CASING AND SPEED RING

The turbine casing and speed ring are cast integrally and are made of cast steel and designed for a hydrostatic pressure of 300 lb per sq in. The casing and speed ring are built in four sections. All connections between sections are flanged bolted joints of ample strength and rigidity.

Each section of the casing is provided with lugs for attaching slings for handling. A number of machined pads for the application of jacks and a corresponding number of 15-ton jacks for leveling the case during erection and supporting it in proper position while the concrete surrounding the casing is being placed are provided.

The design of the casing will permit the removal of all internal parts of the turbine from above. The speed ring is designed to support the weight of the superimposed structure with the casing empty and also to resist the bursting stresses in the casing when subjected to the maximum internal pressure and with no superimposed weight on top of the casing.

COVERS

The top and bottom covers or crown and curbs are made of cast steel, sectionalized to facilitate shipment and handling. A stuffing box designed so that it can be repacked and adjusted without disturbing the guide bearing is provided where the main shaft passes through the top cover. Stuffing boxes are also provided where each wicket-gate stem passes through the top cover. The stuffing boxes will be packed with metallic packing.

The seats for the stationary wearing ring on the turbine top and bottom covers will be ground in the field to their final dimensions, thereby insuring that they will be true circles concentric with the vertical center line of the unit. Sufficient excess metal is allowed in the shop finish of these seats to permit a reasonable variation in the alignment of the centers of the turbines. The grinding of the seats will be done after the casing is completely assembled and embedded in the concrete foundation of the turbine and the generator guide bearings are erected and aligned.

Provisions have been made for checking the clearance between the movable and stationary wearing rings at four points through suitable feeler holes in the top and bottom turbine covers.

GATES AND OPERATING MECHANISM

The gates and gate stems are made of low-chrome-vanadium-manganese cast steel with each gate and its gate stem cast in one piece. Consideration was given to the use of stainless steel for the gate and stem to reduce the rate of wear due to rusting of the upper and lower contact surfaces. The advantages this material offers when compared with alloy cast steel did not justify the additional investment involved.

When the wicket gates are pinched shut under normal operating conditions each gate is subjected to a system of forces such as is shown in Fig. 9. These forces cause bending about both horizontal and vertical axes.

The condition existing in the gates is then one of bending about two axes at right angles to each other. The stress system under such conditions has been discussed by H. M. Westergaard.[6] A chart based on the allowable working stress in simple tension or compression may be constructed as shown in Fig. 9. Using the values of stresses in two directions at right angles as coordinates, a point on this chart is located. If this point lies within the enclosed area, the stress is allowable.

[6] *Journal of the Franklin Institute*, May, 1920.

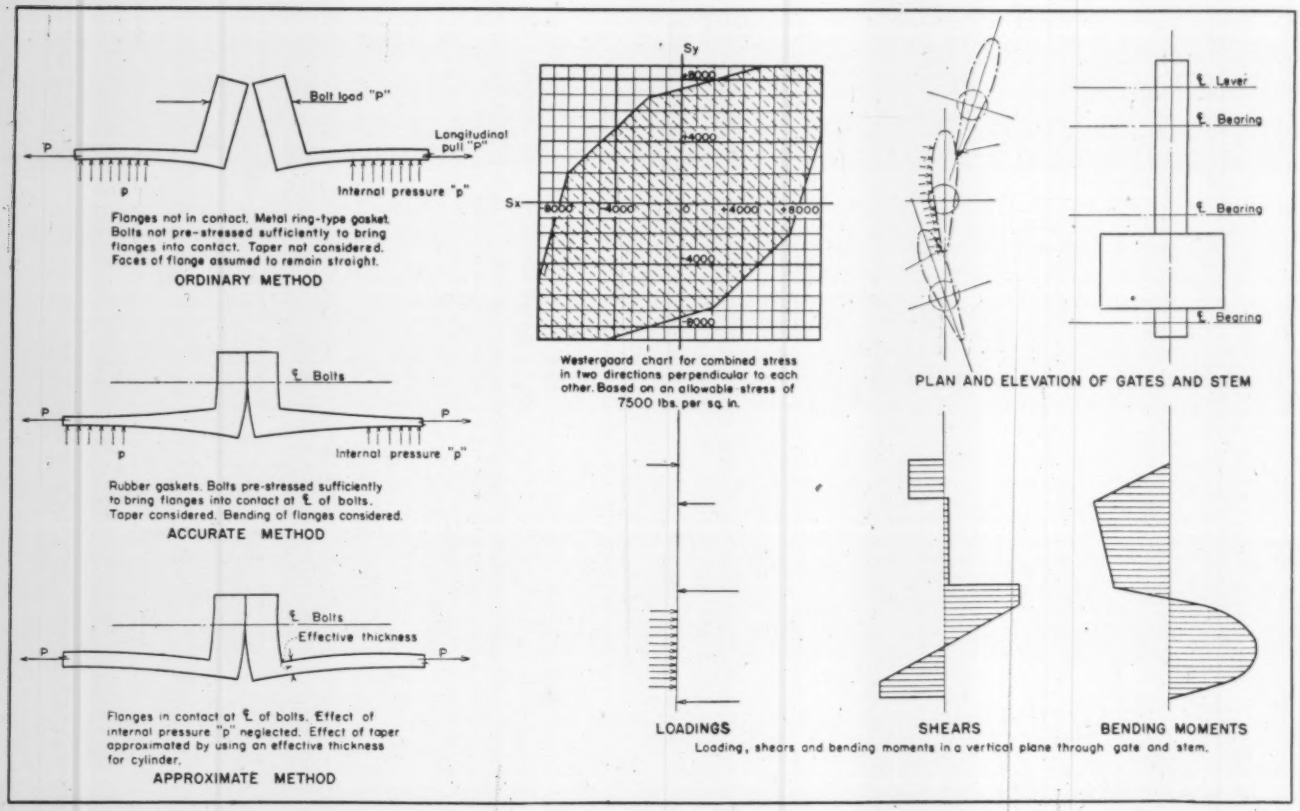

Bolt load "P"
Longitudinal pull "P"
P
Internal pressure "p"
Flanges not in contact. Metal ring-type gasket. Bolts not pre-stressed sufficiently to bring flanges into contact. Taper not considered. Faces of flange assumed to remain straight.
ORDINARY METHOD
℄ Bolts
Internal pressure "p"
Rubber gaskets. Bolts pre-stressed sufficiently to bring flanges into contact at ℄ of bolts. Taper considered. Bending of flanges considered.
ACCURATE METHOD
℄ Bolts
Effective thickness
Flanges in contact at ℄ of bolts. Effect of internal pressure "p" neglected. Effect of taper approximated by using an effective thickness for cylinder.
APPROXIMATE METHOD
Sy
Sx
+8000
+4000
0
-4000
-8000
Westergaard chart for combined stress in two directions perpendicular to each other. Based on an allowable stress of 7500 lbs. per sq. in.
℄ Lever
℄ Bearing
℄ Bearing
℄ Bearing
PLAN AND ELEVATION OF GATES AND STEM
LOADINGS
SHEARS
BENDING MOMENTS
Loading, shears and bending moments in a vertical plane through gate and stem.

FIG. 9 CHECK STRESS ANALYSIS FOR TURBINE FLANGES AND GATES

A similar process may be carried through for emergency conditions when an obstruction is assumed to be caught between adjacent gates. The load applied to the stem by the gate lever is then the limiting value allowed by the breaking link. A Westergaard chart based on the allowable stresses under emergency conditions may be constructed for this case.

The gate stem is subjected to torsion and shear as well as bending about a horizontal axis. The principal stresses in the stem may be obtained from a well-known formula (for combining direct and shear stresses in a plane) and these values may be applied to the suitable Westergaard chart.

Each gate stem is provided with three bronze-bushed grease-lubricated guide bearings, one located in the lower cover or curb plate and the other two located in the top cover or crown plate, one below and the other above the stuffing box. Each gate stem is also provided with a bronze, ring-type thrust bearing arranged to take thrust either upward or downward for supporting the gate in mid-position between the curb and crown plate. Provisions are made for supplying grease under pressure to each of the gate-stem bearings.

A shearing-pin type of breaking link is provided between each gate stem and the gate shifting rings which will be strong enough to withstand the maximum operating forces but will break or yield and protect the rest of the mechanism from injury in case one or more of the gates becomes locked. The pins are designed to fail under double shear and are vee-grooved at the shear plane to reduce bending of the pin to facilitate the removal of the broken parts.

The design of the gates is such that in case any individual gate becomes disconnected from the gate-shifting mechanism, no part of the gate can come in contact with the turbine runner. The entire operating mechanism and the connections for controlling the wicket gates are mounted inside the turbine pit and are readily accessible for inspection, adjustment, and repairs.

The gate-shifting ring is made of cast steel of rigid design and is guided by renewable bronze guide strips. The pull-rod connection from the pressure-regulator valve is located on the shifting ring immediately opposite the adjacent servo-motor piston-rod connection. This connection arm is designed sufficiently strong to stall the servo-motors with 300 lb oil pressure in the cylinders. This feature introduces bending moments not ordinarily involved in a shifting-ring design.

WEARING RINGS AND PLATES

Careful consideration was given to the proper design and material to be used for all wearing surfaces subject to the high-velocity water in the turbine passages. It was decided that two straight parallel seals would be superior to the labyrinth type of seal ring frequently placed on the upper and lower runner and stationary rings. It appeared that a satisfactory job could be obtained by using a soft metal for the stationary wearing rings with a tougher material for the rotating parts. The use of two metals of different characteristics is desirable in the event of accidental contact between the rotating and stationary rings.

The stationary seal rings are of stainless steel, made in segments of a ring and bolted to the stationary seats. The runner seals are of carbon steel forged into a ring and shrunk on the runner bands. It was necessary to reverse the types of material in the two rings, as a satisfactory job of forging the stainless steel into a continuous band could not be obtained. Hard spots developed at the welds that could not be reduced to the desired degree of softness.

The top and bottom distributor wearing plates are renew-

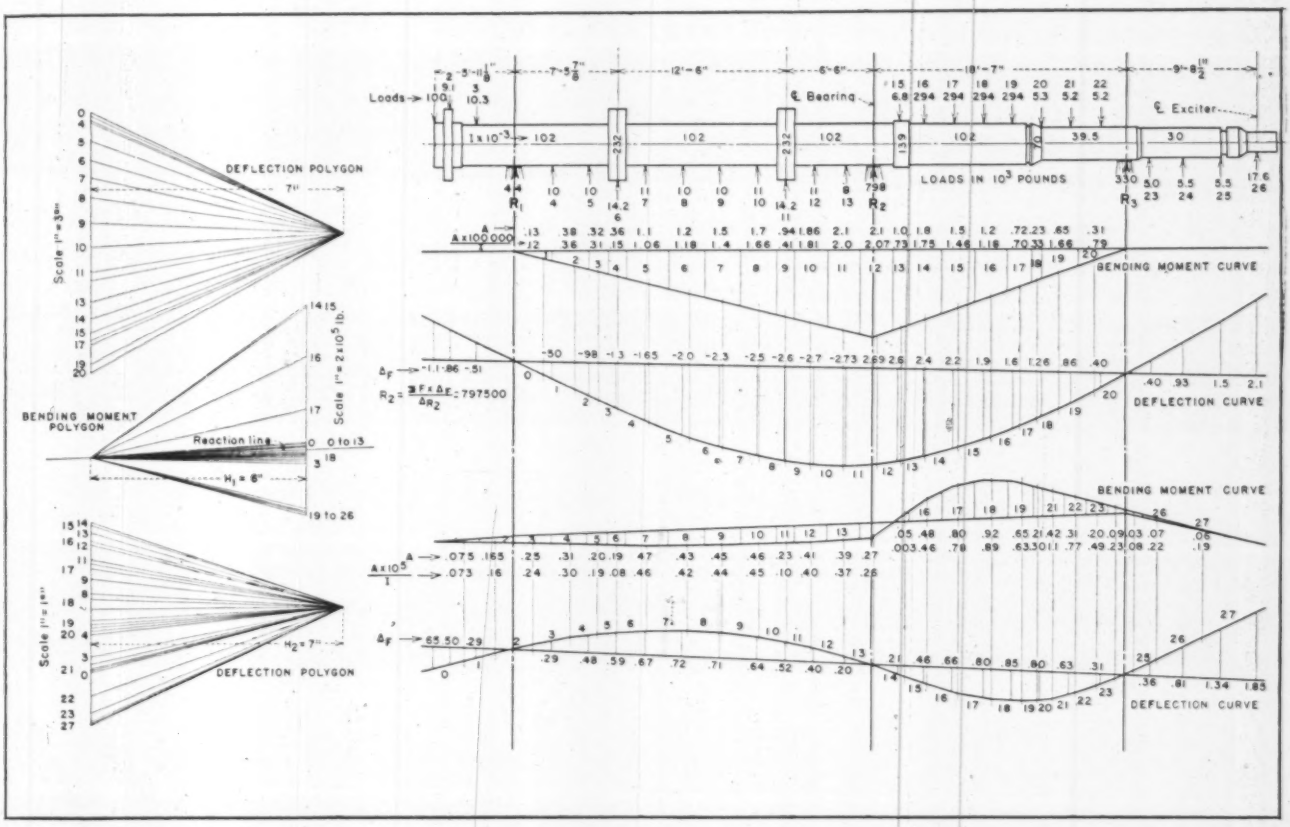


FIG. 10 GRAPHICAL DIAGRAM FOR TURBINE SHAFT

able and are made of chrome-vanadium steel. The runner-seal clearance is 0.060 in. and the distributor-plate clearance 0.015 in.

SERVO-MOTORS

Each turbine is provided with two oil-pressure-operated double-acting hydraulic cylinders or servo-motors having a combined capacity sufficient to exert a torque in the shifting ring of 2,500,000 ft-lb. (See Fig. 14.) The normal operating oil pressure will vary between 250 and 300 lb per sq in. At the oil pressure for which they were designed and with an adequate supply of oil, the servo-motors are capable of moving the turbine gates and pressure-regulator valve a full-opening or a full-closing stroke in 4 sec.

The servo-motor cylinders are so located that the force for moving the turbine wicket gates is approximately divided between the two cylinders and applied in substantially equal magnitude on opposite sides of the gate-shifting ring. The cylinders are mounted on machined surfaces on the turbine casing.

The pistons of the servo-motors are made of cast iron and are fitted with four cast-iron piston rings. The servo-motors are equipped with adjustable ports whereby the rate of closure may be retarded from slightly below the speed-no-load position for maximum head to the fully closed position so as to reduce the pressure rise in the penstock and to cushion the stop just before the guide vanes come in contact.

DRAFT TUBES

A 10-ft section of cast-steel draft-tube liner is provided immediately below the runner. This liner is conical in shape and of rugged design as it forms an important part of the turbine foundation. This casting is provided with two rectangular manholes, 24 in. by 30 in., to permit access to the under part of the turbine runner. Two inspection chambers are provided on the outside of the liner to permit access to the lower runner seals for calibrating the runner clearance at approximately quarter points on the runner bands. A flanged bolted connection is provided on the bottom for matching the $^3/_4$-in. plate-steel liner of the elbow.

The two upper joints of the plate-steel liner are fitted and adjusted in the field after the turbine case has been placed in position. The plate-steel elbow liner is of heavy construction and rigidly braced by means of 4-in. by 5-in. T-beams welded to the outer surface. The draft-tube pier nose is formed in plate steel and divides the liner into two horizontal passages after the 90-deg turn has been completed. The liner will be completely assembled in the shop (see Fig. 11) and match-marked for ready assembly in the field and will be shipped completely knocked down.

The draft-tube liner will be welded in the field. The plates will be held in position by means of bolted butt straps. The butt straps in the roof of the draft-tube liner will be on the inside of the tube, in order that no overhead welding will be necessary. After the liner has been completely welded the inside butt straps will be removed to give a smooth channel for the passage of the turbine discharge water. The bottom horizontal plates are provided with a suitable number of grout holes for the purpose of eliminating entrapped air while the concrete is being poured in contact with the underside of the plates. The entire liner will be held rigidly in position by means of turnbuckles and rods fastened to anchor bolts placed in the first stage of concrete construction.

The turbine draft tube will be unwatered by means of two

water-jet ejectors capable of delivering a combined suction discharge of 3 cu ft of leakage water per sec. Operating water will be supplied from the turbine penstocks. Use of water-jet ejectors for this purpose is very desirable, as they are capable of handling air, sand, gravel, or other debris without injury to the machinery or losing the suction prime, which

FIG. 11 PLATE-STEEL DRAFT-TUBE LINER ASSEMBLED IN SHOP

are troubles commonly experienced when centrifugal pumps are used for this purpose.

PRESSURE REGULATOR

Each turbine will be provided with a pressure regulator or relief valve with a discharge capacity equal to 80 per cent of the turbine capacity. The pressure-regulator valve will be mechanically connected to the turbine-gate operating mechanism with a connection of sufficient strength to stall the servo-motor if the pressure-regulator valve sticks and fails to open. The pressure-regulator valve will be controlled by the governor through the gate-shifting mechanism.

Consideration was given to the use of a pilot-valve arrangement for operating the pressure regulator because of the severe stress conditions imposed by the actuating force of the servo-motor cylinders. It was decided, however, that no chance should be taken of a failure of the relief valve to open with the turbine gates closing and the positive connecting feature was included in the design.

When operating at the higher heads, the turbine gate opening will be limited to prevent the turbine output from exceeding the capacity of the generator, namely, 115,000 hp, and provision has been made for adjusting the pressure regulator so that it will discharge at least 80 per cent but not more than 100 per cent of the water rejected by the turbine upon closing of the gates from the opening as limited.

An oil-filled dashpot is interposed between the pressure-regulator valve and its actuating mechanism and is so arranged that the valve will open to full discharge capacity when the turbine gates move at normal speed through a full closing stroke, and then under control of the dashpot the pressure regulator valve will slowly return, at an adjusted rate, to the closed position. This dashpot permits the turbine gate to close slowly without opening the pressure-regulator valve.

The position of the regulator parts when operating as a relief valve will be as shown in Fig. 12 when the turbine gates are open. Now assume an instantaneous loss of load occurs and the gates completely and rapidly close. The dashpot cylinder is moved toward the right, causing the control-valve stem *A* to be raised by reason of its connection through levers and draw rods to the dashpot piston. There is a slip between this piston and the dashpot-cylinder motions throughout the entire stroke, the amount of which is determined by the setting of the dashpot by-pass valve. This valve must be set to allow the pressure regulator to make its closing stroke at the required rate. The slip of the dashpot piston and cylinder is compensated for in the length of the adjustable lever *B*.

The moment the gates start to close rapidly, the control valve opens and the pressure in the chamber *C* above the main regulator piston *D* is reduced so that it is moved upward (open) by the pressure below the piston acting on the differential area of diameters *K*-1 and *K*-2. This opening process continues as long as the control valve moves away from the main piston with the continued movement of the turbine gates toward the closed position. As soon as the gates are completely closed, the control valve ceases to move upward, and the main regulator piston, continuing its upward movement, almost immediately reduces the control-valve openings, so that a hydraulic balance is reached between the total pressure above and the unbalanced pressure below the piston. The pressure in chamber *E* now forces the control valve toward its seat, reducing the discharge through it and causing the pressure above the main piston to increase and force it down. This action continues until the regulator is closed. Pressure inlet valve *F* is adjusted to provide sufficient water to chamber *C* to force the

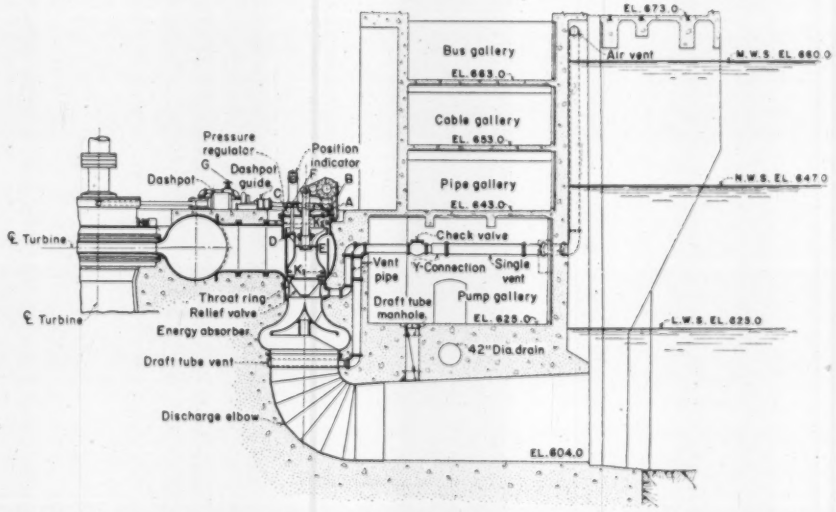


FIG. 12 CROSS-SECTION THROUGH PRESSURE REGULATOR FOR 115,000-HP TURBINE

piston *D* down at the rate of 4 sec for the full stroke. The downward movement of the control valve involves overcoming the resistance of the oil dashpot and it is through this resistance that the rate of closure of the regulator is controlled by means of a valve *G* in the by-pass.

The chart, Fig. 13, shows typical discharge curves of the turbine and the pressure regulator corresponding to the operation just described with full load thrown off the turbine. The chart is made up on the basis of 4-sec time for a full governor stroke in the closing direction, a complete opening stroke of the pressure regulator in the same time, and the gradual moving of the regulator piston from the wide-open to the full-closed position in 36 sec. Assuming that a complete closing move-

ment of the governor and gates in 20 sec would be required if there were no pressure regulators on these turbines, the excess water wasted is shown by the shaded area. The combined discharge of the turbine and regulator, when the latter is operating as a synchronous by-pass, is also shown.

When operating as a synchronous by-pass, the regulator control valve is directly connected to the gate-operating mechanism by a mechanical lock which eliminates the dashpot action.

All movable parts of the pressure-regulator valve are so designed as to be removable for inspection or repairs from above the valve body. The valve body and all parts of the valve and its connection to the turbine casing, which are subject to internal water pressure, are made of cast steel and designed for a hydrostatic pressure of 300 lb per sq in.

The stationary and movable seats of the valve are made of forged steel and are removable. The wearing surfaces are nitrided to give a tough surface to reduce the erosive action of the high-pressure water to a minimum. A hydraulic balancing cylinder is provided in the top of the valve to reduce the force required to operate the pressure regulator. Water pressure is provided by a 12-in. pipe connection leading to the penstock.

Means are provided for hand operation of the pressure-regulator valve with no water in the turbine casing. A recording pressure and motion indicator is mounted on the body of the valve to record and indicate the pressure rise in the scroll case and the position of the movable members of the valve. These records will enable the operator to make the necessary adjustments to obtain the desired regulation.

The pressure-regulator discharge valve discharges into a through-type energy absorber placed immediately below the valve orifice. The energy absorber is made of cast steel and is provided with a nitrided wearing surface to reduce to a minimum the erosive action of the discharge water. The energy-

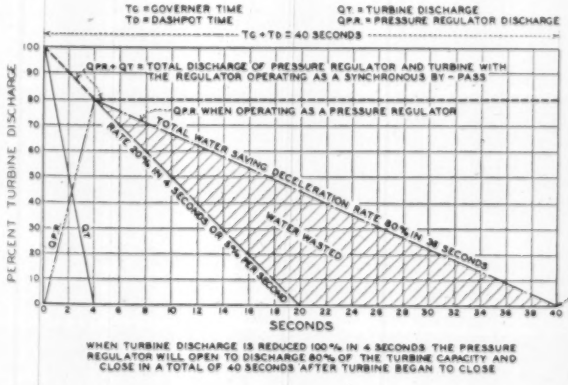

TG = GOVERNER TIME
TD = DASHPOT TIME
QT = TURBINE DISCHARGE
QPR = PRESSURE REGULATOR DISCHARGE
TG + TD = 40 SECONDS
QPR + QT = TOTAL DISCHARGE OF PRESSURE REGULATOR AND TURBINE WITH THE REGULATOR OPERATING AS A SYNCHRONOUS BY-PASS
QPR WHEN OPERATING AS A PRESSURE REGULATOR
TOTAL WATER SAVING DECELERATION RATE 80% IN 36 SECONDS
RATE 80% IN 4 SECONDS OR 20% PER SECOND
WATER WASTED
QT = 0
QPR = 0
PERCENT TURBINE DISCHARGE
SECONDS
WHEN TURBINE DISCHARGE IS REDUCED 100% IN 4 SECONDS THE PRESSURE REGULATOR WILL OPEN TO DISCHARGE 80% OF THE TURBINE CAPACITY AND CLOSE IN A TOTAL OF 40 SECONDS AFTER TURBINE BEGAN TO CLOSE

FIG. 13 CURVE SHOWING EXPECTED DISCHARGE CHARACTERISTICS FOR PRESSURE REGULATOR FOR 115,000-HP TURBINE

absorber casting is heavily ribbed to provide a solid anchorage in the concrete.

The elbow of the draft-tube section is lined with plate steel 3/4 in. thick, heavily ribbed and anchored into the concrete foundation, and is of the same type of welded construction as described for the turbine draft-tube liner. Consideration was given to the desirability or necessity of lining the horizontal discharge portion of the draft tube. It was decided that by providing ample means for aerating and relieving the pressure within the elbow, no harmful effects would result from the relatively high discharge velocities alone.

Extensive tests were run by S. Morgan Smith Company in their hydraulic laboratory to determine the correct design of many features of the pressure regulator, and data on the operating characteristics were obtained. These tests were made on a model with an inlet 6 in. in diameter, one-fourteenth of the size of the Boulder dam regulators. The test head available was from 70 to 120 ft. The testing program included the trial of several sizes and shapes of throat ring and valve seat, selec-

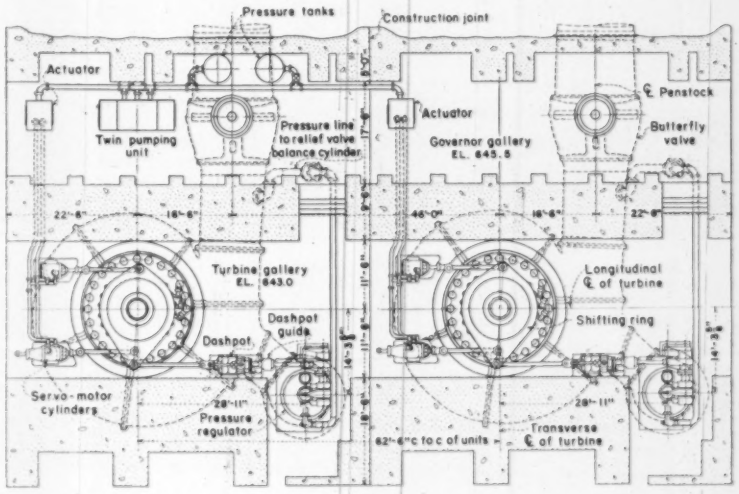


FIG. 14 ARRANGEMENT OF TURBINES, PRESSURE REGULATORS, AND GOVERNING UNITS FOR 115,000-HP TURBINES

tion of draft tube, determination of the opening required in the energy absorber, and the best design of the main regulator piston from the hydraulic standpoint. During these tests, and as a main part of the program, the operating characteristics of the valve were carefully determined with particular attention to the forces on the main piston and on the control valve throughout the opening and closing strokes.

The conditions under which the pressure regulator will be required to operate are somewhat different from the usual installation. This is due to the fact that the discharge passage will be 47 ft below tailwater elevation and the air-relief lines are initially filled with water. For this reason careful consideration had to be given to the problem of aerating the low-pressure points in the discharge portion.

An analysis was made to determine the proper size of vent pipes by assuming that the air would be entrained by the high-velocity jet in a manner similar to that of the Taylor air compressor. It was assumed that the amount of free air that could be entrained at these points would be equivalent in volume to 50 per cent of the water discharged. It was further decided that a pressure of approximately one-half of an atmosphere would be the allowable minimum to prevent cavitation of the steel liner. These studies indicated that a 12-in. pipe would be sufficient for the admission of air to the throat rings.

As the jet discharge from the lip of the energy absorber is toward the center of the discharge elbow, it was thought essential that means should be provided for the admission of air immediately below the discharge lip. It was estimated that a 14-in. pipe would be required at this point to lower the pressure to approximately one-half of an atmosphere. The 14-in. air line is provided with a check valve to prevent a short-circuit of flow between the draft-tube and throat-ring vents in case there should be an unbalanced pressure between these two points

under high-tailwater conditions. This check valve could have been eliminated were it practicable to carry both atmospheric relief outlets above maximum high tailwater.

GOVERNORS

The governors for the 115,000-hp turbines are of the oil-pressure, relay-valve, actuator type with electrically driven speed-responsive element and are so designed as to permit the operation of the unit at either 50 or 60 cycles with a minimum change in the apparatus. The speed-responsive element is motor driven from an alternating-current generator with permanent magnets, located at the top of the generator shaft.

The speed change from no load to full load is adjustable from 0 to 6 per cent. The governing system is designed to operate with an oil pressure ranging from 250 to 300 lb per sq in. The capacity of the system is such that the turbine gates can be closed from full-open to closed position in 4 sec, with maximum head on the plant.

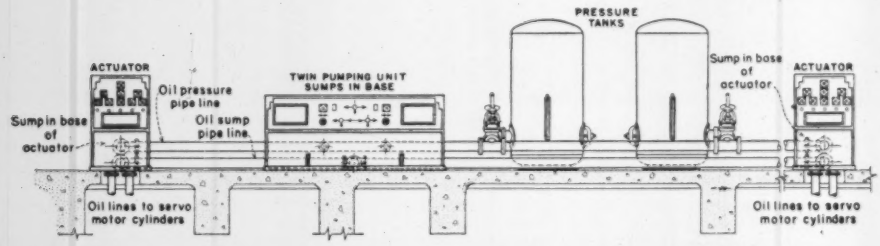

PRESSURE TANKS
ACTUATOR
Oil pressure pipe line
TWIN PUMPING UNIT SUMPS IN BASE
Sump in base of actuator
Oil sump pipe line
Sump in base of actuator
ACTUATOR
Oil lines to servo motor cylinders
Oil lines to servo motor cylinders

FIG. 15 PUMPING-UNIT ASSEMBLY AND GOVERNOR ACTUATOR FOR TWO 115,000-HP UNITS

The actuators and auxiliary-governor equipment are located on the main turbine floor at elevation 645.5. (See Figs. 14 and 15.) The speed-responsive elements are capable of causing the actuator to effect a corrective movement of the turbine gate in consequence of a speed variation of its turbine of one one-hundredth of one per cent or less. The speed of the responsive elements will vary directly and in exact proportion to the speed of the main shaft of the turbine for all rates of acceleration and deceleration, including starting. The governors will automatically close the turbine gate at the normal rate of closure in case of failure of the governor drive.

The following special features are incorporated in the actuator designs:

(*a*) A load-limit device operated manually at the actuator or electrically by remote control from the switch board.

(*b*) A device for controlling the speed of the turbine, which can be operated manually at the actuator, and also electrically operated by remote control from the switchboard.

(*c*) A device for closing the turbine gate at normal rate of closure, which can be operated manually at the actuator and also electrically operated by remote control from the switchboard and also from the main generator floor.

(*d*) A device for automatically closing the turbine gate and giving an alarm upon failure of the governor drive, failure of governor oil pressure, excessive temperature of turbine guide bearings, or upon operation of any of the automatic protective features in connection with the main generator, exciter, pilot exciter, or transformers.

(*e*) A device for automatically matching the speed of the turbines with the speed of the system to which the unit is to be synchronized.

(*f*) Two electrically operated speed indicators, one mounted on the actuator column and one mounted on the switchboard. These speed indicators are driven from a magneto-type generator driven from the main shaft of the turbine.

(*g*) An automatic air valve for controlling the operation of the generator brakes. This air valve will be capable of applying the brakes on and off at intervals permitting rapid braking without excessive heating of the brake shoes and rings.

(*h*) A steel-tape restoring mechanism operating in a closed conduit and over sheaves at turns, thereby eliminating all lost motion between the servo-motor piston travel and the governor pilot valve. The pilot-valve arm is spring loaded to maintain substantially a uniform tension in the restoring tape.

Of special interest from the standpoint of appearance are the casing and instruments designed for the actuator stand and pumping-unit housing, illustrated in Fig. 15. This type of cabinet design should be particularly pleasing where it is located on the generator floor.

The governor-oil pumping system is designed to operate two actuators as shown in Fig. 14. Each pumping assembly is provided with two motor-driven high-speed gear-type oil pumps. Each pump will have a capacity equal to two and one-half times the volume of the servo-motor cylinder of one turbine per minute. The pumps are interconnected for automatic control so that either pumping unit may be used for normal operation with the other unit serving as a stand-by. The motors are arranged to start automatically on failure of the electric current supply to the operating pump or when the oil pressure or level falls a predetermined amount below that pressure or level which starts the supply of oil flowing from the operating pump.

For each governor a pressure tank is located in an alcove near the actuator. The pressure tank has sufficient capacity to supply two complete servo-motor strokes of the turbine with a drop in pressure from 300 to 250 lb per sq in. without the aid of any pumping. Float valves are provided in the bottom of the pressure tank to close automatically in case the oil level drops to prevent air from the pressure tank from entering the governing system.

An unusual feature of this governor installation is the elimination of oil sumps below the governor floor. This feature of design was effected by converting the bottom portion of the actuator and pumping-unit bases into the sump space. This arrangement is made possible because of the relatively small sump space required in a high-pressure system. The housing around the apparatus is of welded plate-steel construction which permits the storage of oil around the outside of the usual cast governor and pumping stands.

ACCEPTANCE TESTS

After the turbines have been installed and placed in satisfactory operation, they will be tested to determine the turbine horsepower, water discharge in cubic feet per second, and horsepower-efficiency performance. The electrical generator will be used as a brake to measure the turbine horsepower output. The water-input measurement will be made by the Gibson time-pressure method, using the differential-pressure diagrams. The water-input measurements will be coordinated by the index method of testing, using suitable differential piezometer taps located in the turbine scroll case.

The Gibson apparatus will be connected to two piezometer rings located in the 13-ft diameter penstocks, with the orfices 100 ft apart. The net pressure head on the turbines will be determined by means of a piezometer ring attached near the

turbine intake, actuating a calibrated pressure gage of the fluid-pressure-scale type which weighs the water pressure on a plunger. The tailwater elevation will be determined by means of suitable float gages located in the tailrace. The windage and friction losses will be determined for both the runner filled with water and the water surface depressed below the runner by means of compressed air, with the turbine gates closed and the generator operating as a motor.

TEST OF GOVERNOR CHARACTERISTICS

The governor characteristics will be tested by means of a multi-tube oscillograph recording on a common photographic negative the functioning of the essential parts of the turbine and governor under various conditions of load change and operation. The performance of the pressure regulator will be tested in a similar manner. No provisions are made for automatic recording of the discharge of the pressure-relief valve.

The relief-valve discharge for various positions will be calibrated by the Gibson test with the turbine gates closed and the relief valve operated by hand control. The amount of accumulative water discharged through the pressure regulator will be computed from the gate-opening record chart.

The scroll-case differential-pressure taps will be connected permanently to a meter register to record and total the discharge of the water through the turbine. This data will be used to check the daily over-all operating efficiency and to coordinate the wear on the turbine with the running load factor on the unit. A special manometer will be provided for connecting to the pressure taps to make turbine efficiency determinations at intervals to anticipate when the turbine should be repaired.

ACKNOWLEDGMENTS

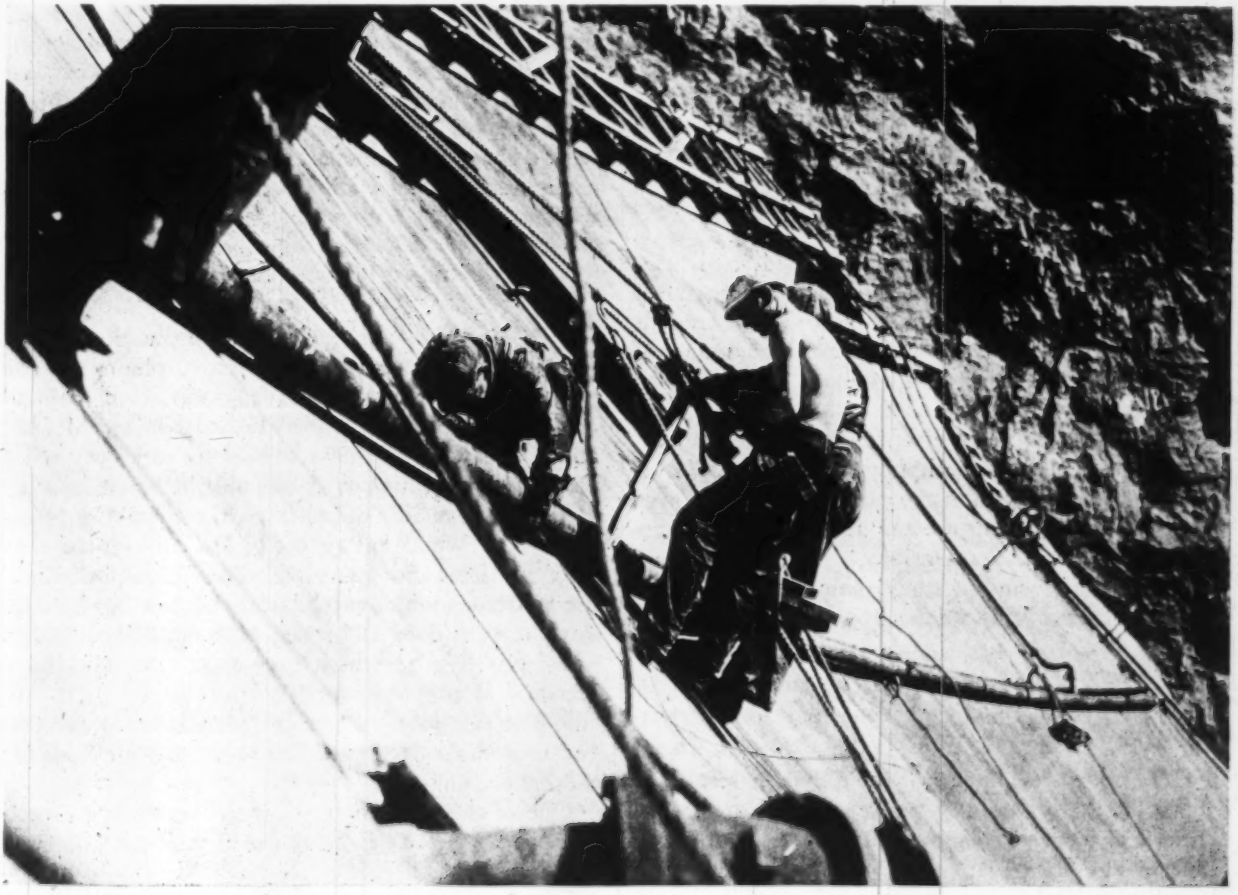
The check analysis for the turbine shaft was contributed by Eugene E. Stoeckly, and the scroll-case flange and turbine-wicket-gate stress analyses were contributed by Douglas C. Seeley, both of the engineering forces of the Bureau of Reclamation.

The pressure-regulator characteristics were contributed by Grant H. Voaden, hydraulic engineer, and George A. Jessop, chief engineer, S. Morgan Smith Company, York, Pa.

The hydraulic turbines are being furnished by the Allis-Chalmers Manufacturing Company, Milwaukee, Wis.; the pressure regulators by the S. Morgan Smith Company, York, Pa.; and the governing equipment by the Woodward Governor Company, Rockford, Ill.

The Boulder power-plant designs were prepared under the direction of J. L. Savage, chief designing engineer, and L. N. McClellan, chief electrical engineer, of the Bureau of Reclamation, with the author in responsible charge for the design of the mechanical installation.

The engineering and construction work for the Boulder Canyon project is under the general direction of R. F. Walter, chief engineer, Bureau of Reclamation, with headquarters at Denver, Colo., and all activities of the bureau are under the general charge of Dr. Elwood Mead, Commissioner, Bureau of Reclamation, Washington, D. C.



PIPEFITTERS WORKING ON THE DOWNSTREAM FACE OF BOULDER DAM

MECHANICAL ENGINEERING

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GEORGE A. STETSON, *Editor*

Light-Weight Welded Structures

FREQUENTLY mentioned among the reasons for believing that engineering skill has not yet run itself out but is still able to provide new methods and new projects are the high-speed, light-weight, streamlined trains with which the railroads are hoping to win back passengers now making use of other forms of transportation. The progressive A.S.M.E. Railroad Division, at the Semi-Annual Meeting in Denver, is providing three papers on this subject, two of which describe the Burlington *Zephyr* and the third the Union Pacific's light-weight trains, the first of which was described in the December, 1933, issue of MECHANICAL ENGINEERING.

There is a significance to all engineers in the papers on the Burlington *Zephyr*, for it takes little stretch of the imagination to see that the kind of construction used can be applied to other than railway-car structures. The numerous objectives which the engineers of the Budd Manufacturing Company and the Burlington railroad attempted to attain in the *Zephyr* included a strong, light-weight, corrosion-resistant construction, and this involved, among other things, a metal of suitable characteristics, a welding process that could be controlled with precision, and an accurate knowledge of the stresses which the structure must of necessity resist. It meant a judicious combination of new with old and tried principles and methods of design and construction. It called for an ability to compute where the least amount of material would provide the greatest amount of strength because "factors of ignorance" could not be tolerated.

The papers by E. C. Anderson and R. Eksergian describing the train and the design problems involved in it and analyzing the stresses in the car structure have been preprinted for the Denver meeting and are shortly to appear in the A.S.M.E. Transactions. All forward-looking engineers will want to study them to find out how the principles involved can be applied to their own problems.

The Boulder Dam Project

MUCH water has flowed past the dam site since Dr. W. F. Durand, past-president, A.S.M.E., presented a paper on "The Problem of the Colorado River," in the February, 1925, issue of MECHANICAL ENGINEERING. Practically the entire issue this month is devoted to papers discussing certain engineering details of the power project that has been under development at Boulder dam, one of the phases of the problem discussed by Dr. Durand.

Dr. Durand's paper outlined the four major aspects of the problem of the Colorado River to which the Boulder dam project is the solution: (1) Removal of the flood menace to the Imperial Valley in California, (2) the need for an abundant water supply in the Southwest for municipal and irrigation purposes, (3) the choice of a dam and reservoir site, and (4) the utilization of the energy of the water at the dam site in the generation of electric power for which an ever-increasing demand exists in the Southwest.

Disputes loud and bitter have raged over the project and some of its numerous features. Time alone will prove the wisdom or the folly of the undertaking. Man once more has set himself the task of altering the face of Nature. He must abide by the consequences; for where he solves one set of problems he raises another which will challenge at some future time his ability to cope with them.

The present concern of engineers is with the engineering features of the project, and here, as demonstrated by the papers in this issue describing the work, there is much that will arouse enthusiasm and pride in the members of the profession who planned it and carried it out and in the industries of the country that provided the expert services and the materials and equipment without which it cannot be successful. The A.S.M.E. is grateful to the Commissioner of the Bureau of Reclamation for permission to publish in MECHANICAL ENGINEERING the papers its engineers have prepared describing the project, and is highly appreciative of the magnanimity with which these engineers acknowledge their debt to the engineers and industrial concerns that collaborated in the work.

Mechanical Engineering at Boulder Dam

THOSE who imagine that the building of great reclamation and hydroelectric plants is chiefly a matter of interest to civil engineers will be amazed on reading the papers by Messrs. Kinzie, McClellan, and Winter, and the companion paper by Messrs. Day and Bier which will be published in a later issue, at the great variety of mechanical engineering problems involved. We all take more or less for granted the fact that turbines and generators and their appurtenances are matters in which mechanical engineers are concerned, but the penstocks and gates at Boulder dam also provided problems in which they have very immediate interests. While the work of installation calls for an immense amount of reinforced-concrete construction, the equipment demanded the shops and methods of the mechanical engineer.

One of the most interesting and significant aspects of the project is the extensive use of fusion welding. The paper by P. A. Kinzie in this issue strikingly illustrates this point. One unfamiliar with the rapid development of welding in the last few years will be amazed at the

satisfactory manner in which this method of construction has been substituted for casting in steel, and at the precision with which large and fairly complicated structures can be built up out of welded steel plates.

A Shortage of Engineers?

SPEAKING over the radio on the evening of May 31, Dr. Harvey N. Davis, president of Stevens Institute of Technology, startled his hearers by saying that "four years from now there won't be enough engineering graduates to meet even the technical demands of the profession, to say nothing of all the other opportunities for constructive leadership that lie ahead."

To engineers who have been without employment for many months, to recent graduates who have found no opportunity in engineering since their alma maters turned them loose upon the world, to members of the graduating classes not yet absorbed in industry, and to boys and fathers of boys who are faced with the problem of seeking an education as a background for what has frequently been termed an overcrowded profession this courageous and optimistic statement conveyed a variety of degrees of doubt and hope. Did an over-enthusiastic educator permit his zeal to run ahead of his judgment, or is there reason to believe that the statement is true? Opinions on this will differ, but there are telling arguments in favor of Dr. Davis' diagnosis.

Dr. Davis himself backs up his statement by citing three principal reasons. (1) The graphical analysis of engineering graduates from all the schools of the country from 1920 to 1937, prepared by Gen. R. I. Rees, of the American Telephone and Telegraph Company, showed that the number is decidedly below normal for the years 1935, 1936, and 1937, so that by 1937 these schools will be graduating something like two-thirds of the men indicated as necessary by the normal-trend line of the chart. (2) James A. Farrell has been quoted as saying that American industry would soon find itself facing the worst shortage of technically trained men it had ever known in all occupational levels from skilled workmen to research workers with doctor's degrees. Such a shortage is a matter of record in post-depression industrial history. (3) The scope and range of the engineering field is continually broadening, research has been actively advancing during the depression, and already new fields are opening up.

Other testimony than that which Dr. Davis cited can be brought to the defense of his thesis. There is, for example, the prevailing experience that 1932 was the low ebb of jobs for graduating engineers, that 1933 was better, and 1934 is still better. The economic cartographers call attention to the curves of industrial production in this country which, eliminating the violent oscillations marking the change of national administration, show a general rise from the low point of 1932. Indexes of production in other countries, with but few exceptions, show the same tendency to rise, and this is confirmed, historically, by the long-time trends that stretch back for more than a century. While the situation is far from being bright and safe when other factors are considered it does not serve the present purpose to dwell upon these very real and alarming hazards. The evidence is such that both fatalist and optimist can agree with Dr. Davis. We are headed for better times.

Preparing for Better Times

ASSUMING that Dr. Davis, the optimists, and the fatalists are right and that we are facing the prospect of better times and a shortage of good men for technical and engineering positions, the question arises, What plans are being made to improve the quality of the engineering profession?

Dr. Davis, in his radio broadcast of May 31, gave evidence to show that certain engineering schools are alive to what demands on the profession will mean to them. Guidance at a very early stage of a man's career is receiving an increasing amount of attention. Readers of MECHANICAL ENGINEERING will probably recall the work that has been going on under Dr. Davis' direction at Stevens in the Junior Camp for secondary-school boys, the object of which is to make sure that boys who choose to enter an engineering school are mentally and temperamentally fitted to profit by it. The twofold object of encouraging the fit and eliminating the unfit benefits the boy and makes for a better engineering profession. In various orientation courses throughout the country, similar objectives are aimed at, and this spring there was announced an engineering conference for boys at Lafayette College, which commences about the time this issue of MECHANICAL ENGINEERING is placed in the mail. In the terms of the prospectus the conference will be devoted to "the vital problem of turning toward the engineering schools only those who have the high qualities, interests, and capacity required to obtain intellectual satisfaction from the pursuit of one of the engineering courses." In the quotation the word "only" warrants special emphasis.

Not alone the schools, but the profession is preparing to maintain high standards of quality and accomplishment in the future. The Engineers Council for Professional Development, whose objectives have been frequently explained in these pages. The enthusiasm with which they have been greeted is adequate proof of the fact that engineers have the desire to improve their profession and prepare it for the greater service the future has in store for it.

These ideals and the mechanisms set up to approach them are only a step, important and significant as it may be. It is to be hoped that out of the experiences of the recent years there will be born a livelier consciousness of what the engineering profession may accomplish in the way of technological progress and of the manner in which this progress may be more mercifully integrated into the lives of common folk. There is much to be done, and it is not too early to begin in the secondary school to pick the best men for service in the engineering profession in the better times to come.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

STRENGTH OF MATERIALS

THIS is a symposium on the practical applications of recent research into the strength of materials, based on the papers presented at a meeting of the Verein deutscher Ingenieure in November, 1933, in Essen. Only certain parts of this very interesting matter can be abstracted here.

CORROSION STRESS AS CAUSE OF RUPTURE

Observations have shown that in the majority of cases a material decrease in long-range resistance to stresses occurs in connecting rods of double-acting two-stroke-cycle engines due to the effect of the cooling water. Extensive tests have been undertaken by the M.A.N. Company to determine the properties of various materials of construction in this respect. All the samples were submitted to a maximum of 100,000,000 alternations of stress, as it was found that the limit of 10,000,000 alternations formerly used was not sufficient, and material continued to lose strength when tested after that limit. Tests were made with various corroding liquids from fresh water to sea water. The test pieces were flat bars subjected to from 3000 to 4000 alternations per min on a M.A.N. bending oscillating machine. The results of these tests are given in the form of curves in the original article. Among other things it was found that corrosion was materially greater in the larger bores, that is, at lower velocity of flow of water, than in smaller bores and higher velocity of flow. In some connecting rods where a high water velocity was used, erosion was found, which, however, had no connection with corrosion cracks. Increase in temperature of cooling water produces materially greater corrosion. This was found in the case of a steel pipe where cold cooling water flowed on the inside while the outside was exposed to the action of the return flow of the warm cooling water. While the inner surface showed only slight indications of corrosion, the outer surface was quite strongly eaten away. This leads to a practical conclusion to the effect that the cold water should be made to flow through the drilled holes in connecting rods, while the warm cooling water should be returned through a pipe made of corrosion-resisting material. It is of particular importance to note that in the range of the largest heat stresses, no corrosion was observed. Contrary to the theoretical assumptions this zone is therefore least exposed to the danger of corrosion. Corrosion is assisted by guide strips inserted in the bore-holes. Cracks were frequently observed below and above these guide strips, but these cracks did not appear until after four to five years of operation.

As regards protection against corrosion, the most effective method is to insert into the bore-hole of the connecting rod a thin-walled tube made of a corrosion-resisting material. Protective mediums against corrosion when added to the cooling water have given very good results. Even a small addition of lubricating oil to cooling water produces a material improvement. In general, the effect of corrosion was materially greater in the case of soft water than in hard water, particularly with water containing common salt.

This report is a good example of the practical value of long-range testing. The observations made in the course of the tests show quite clearly that breaks in connecting rods are practically exclusively due to fatigue induced by corrosion and the elimination of troubles from this source can be achieved only by a fundamental study of corrosion fatigue as affecting the strength of materials.

COMPUTATION OF THE STRENGTH OF TURBINE BLADES

The author, Dr. of Engrg. Mellan, considers separately the first and last rows of blades. The first rows are, of course, exposed to the high temperatures of the high-pressure steam. As the blades pass through the jets of steam coming from the nozzles they received periodic impulses, the frequency of which depends on the peripheral velocity, the number of blades, and the number of nozzles. This imposed frequency should not coincide with the natural frequency of vibrations of the blades, as otherwise resonance will be produced. This requirement is, as a rule, satisfied automatically as, for other reasons than that of the frequencies, the blades are short and very resistant against bending. The oscillating stresses induced by the impulses are therefore comparatively low. On the other hand the maintenance of strength requirements is made difficult by the use of high operating temperatures. The long-range strength of such materials as may be used for blades under high temperatures has scarcely been investigated and experimental work in this direction is imperatively necessary. All that can be done today is to draw conclusions from tests on the strength of blades at room temperatures. Furthermore, in this connection the effect of corrosion due to the presence of salts in the steam and the effect of surface conditions, neither of which has been properly studied, must be considered. On the other hand, however, we know how to calculate reliably the tensile stresses induced by centrifugal forces and the bending alternating stresses induced by variations in steam pressure.

The last rows of blades are very long, but, on the other hand, the steam temperature is here only from 30 to 50 C (86 to 122 F) so that the difficulties due to high temperatures do not need to be considered here. The vibrations due to the impingement of steam from the nozzles also need not be considered here as the stationary guide blades are concentrically located about the rotors. The author considers in some detail, however, the creation of vibrations in the blades. In the last rows of blades the problem is to calculate the natural periods of vibration in the blades and to lay out the blades so that their natural periods of vibration should be from two to three times higher than the operating periods of vibration. The results of approximative processes for this kind of calculation, only very recently published, have been confirmed by practical experience.

LONG-RANGE RUPTURES IN MOVING PARTS OF HIGH-SPEED DIESEL ENGINES

The author of this paper, Dr. of Engrg. E. Flatz, of Cologne, raises the question of the methods of calculating stresses in such machine parts as those entering into the construction of

high-speed Diesel engines, and, in particular, the value of long-time tests. He says that results of recent research can be successfully used to create new machine parts and new shapes, and to meet new kinds of stresses, but calculation of actual stresses in motor parts is still quite difficult, as the shape of the part is not easily handled mathematically, and the forces occurring in practical operation can be given only in a very approximate manner, with the result that the results obtained may vary as high as 100 per cent. Moreover, the calculation of stresses requires so much previous experience that it can be made successfully only by highly trained engineers.

In computing standard machine parts differing only slightly from those commonly used the Humboldt-Deutz Motors Co. uses certain simple models. In large organizations this makes it possible to proportion more uniformly more or less similar machine parts. It must be remembered, however, that, while the value of permissible strains assumed in model tests has the dimension of the stress, it does not by any means represent the actual strain. When, however, proper precautions have been taken the method of determining stresses by means of models cannot be said to be opposed to the spirit of the modern science of strength of materials.

The design of a part of a motor as a body of a strength equivalent to that of a model is, however, time consuming and greatly limited by practical considerations, particularly those relative to the methods of manufacture. Research in materials has hitherto been content, particularly in long-range tests with machined pieces or models, to find ways of eliminating troubles after they have occurred. Experience shows that ruptures in motor parts are practically exclusively due to peak stresses produced by disturbances in the "flow of power." For practical purposes, it would appear to be particularly necessary to develop a theory of peak loads that would hang together well. As regards research generally Doctor Flatz makes the following statement: "Even the results of research that we have available today are of inestimable value. While precise computation of strains in motor parts has not been very well developed as yet and has not been carried out far enough so that it could be applied to shapes of machine parts very different from those now used, the economical value of such knowledge as we have on the subject of peak strains is tremendous. Without exaggeration one can say that by this research millions of marks have been saved in the German economic system."

CRANKSHAFTS

In Fig. 1 are plotted for a number of crankshafts of small single-cylinder Diesel engines curves of nominal stresses obtained in accordance with the usual formula for strength of materials for cross-sections *a* and *b*, assuming a bending stress due to the maximum pressure on the piston.

Fig. 1 is based on analytically determined bending strains ("nominal stresses"), *a* in the crankshaft pins and *b* in the crankshaft cheeks.

The broken lines apply to strains in crankshafts of the third group, after reinforcement. No fractures were here observed. The first two ordinates apply to carburetion motors and the next three to Diesel motors. The bending moments were determined exclusively from the ignition pressure on the piston (50 atm gage for Diesel motors and 20 atm for carburetion motors) and distance between bearings, without giving any consideration to forces of acceleration. This has been done, as only comparative figures for motors of the same kind were desired. The various figures apply to German standard steels. *ME2Wv* is a manganese steel.

The results of a large-scale investigation of breakages in crankshafts from among 150,000 motors now in operation shows that when the crankshafts have been built out of the German standard steel St 50.11, breakage occurred when the "nominal stress" at *b* was higher than about 7 kg per sq mm (9954 lb per sq in.) or corresponds to a strain of about 11 kg per sq mm. (15,642 lb per sq in.) at *a*. As the alternating bending strength of this steel is about ± 25 kg per sq mm (35,550 lb per sq in.), it would appear that the actual strain at *b* is about 3.5 times as great as the nominal strain.

From this statistical analysis of breakages, proper indicative values for the selection of permissible nominal stresses to be used in the design of this kind of shaft have been obtained.

CONNECTING-ROD BOLTS

This machine part requires the most earnest consideration, as its failure in the majority of cases results in a complete destruction of the machine. The fracture practically always occurs in the first stressed thread turn. The following have

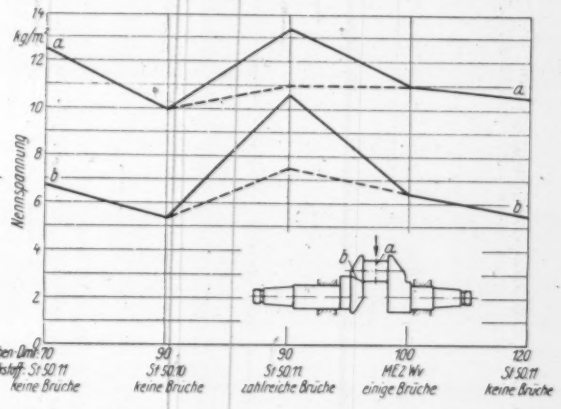

FIG. 1 EVALUATION OF OBSERVATIONS ON 150,000 CRANKSHAFTS OF SMALL MOTORS BUILT BY THE HUMBOLDT-DEUTZ MOTOR CO. (*Nennspannung* = "nominal stresses;" *kolben-Dmr*:70 = diameter of piston in mm; *werkstoff* = material; *keine Brüche* = no cases of breakage; *zahlreiche Brüche* = numerous cases of breakage; *einige Brüche* = some cases of breakage.)

been found to be its causes: (*a*) Failure of material, particularly porosity due to improper heat treatment; (*b*) improperly cut thread, loosening, or crooked seating of the nut. The following are recommended as means to prevent such failure: (1) Employment of materials not too sensitive to heat treatment; (2) the most careful machining of the thread, which should be milled and strengthened by surface pressure treatment; (3) the most careful machining of the nut, with the bearing surface precisely normal to the axis of the screw; (4) provision of the right amount of stressing in erecting and protecting the nut against loosening.

Connecting-rod bolts have hitherto been computed chiefly in respect to tensile strength. As a basis for the calculation there were used the mass forces resulting from the motions of the piston and connecting rod and occurring in a four-stroke-cycle Diesel engine only at the end of the exhaust stroke and in two-stroke-cycle engines only when the compression failed. Even if, for purposes of this kind of calculation, a nominal strain was assumed thirty times the elastic limit, cases of fracture could not be avoided. In the Humboldt-Deutz Motor Co. the bolts are now so computed that they are given a sufficient initial stress to take care of the friction between the two halves of the connecting-rod head, bearing in mind the acceleration of a cross-section of the rod. The fundamental features of this calculation do not depend on the way the motor works. If we consider that the peak loads occur at the base

of the thread, it will be found that if the dimensions have been determined from the erecting stress alone there will be at most a factor of safety of two. Experience has shown that where closure by friction is lacking, particularly when, because of the rotary oscillations of the crankshaft, very large accelerations of cross-section appear, the screws fail. The correctness of the present method of computation has been confirmed by measuring the erection stresses, these measurements having been carried out by calibrating the terminal surfaces of the screw and determining the total elongation of the screw as a measure of the strain.

MATERIAL FOR STEAM BOILERS

The paper by Dr. of Engrg. E. Lupberger deals with the selection of material and wall thickness in steam boilers. Here again the opinion is expressed that the empirical methods now used cannot endure. But in order to replace them by really scientific methods one must above all know the spacial field of stresses at the critical locations in the boiler. They cannot be determined for practical purposes by mathematical methods. The solution of the problem can rather be found only by experiment. It is therefore the intention within a very short time to submit to elongation measurements the few typical parts which alone are of importance for the design of new boilers, and to determine in this way the distribution of stresses. These experiments are to be carried out as a community undertaking and the results are to be presented in a "Book of Types." The purpose of the new development is the creation of bodies that are as smooth as possible and have undisturbed flow of stresses and as uniform a distribution of strains as possible. In boiler construction only a few simple elements are really needed.

Measurement of strains can be carried out on a boiler only at static pressure and as a rule at temperatures of about 20 C (68 F). In boiler operation, however, alternating forces are superimposed on static loads. Variations of steam pressure are of the very smallest importance as a source for the appearance of alternating strains, as the pressure variations in boilers while in operation seldom exceed ±5 per cent. The alternating stresses due to variations of temperature are much more important. This view is supported by the observation that in old boilers the formation of cracks is greatly accelerated by the installation of elastic firing. Therefore, in the design of boilers particular attention should be paid to the mobility of individual parts, such as may be due to the changes of temperature and to stresses generated in that way.

As regards the investigation that it is planned to undertake, it is stated that the classical science of strength of materials avoided the uncertainty as to the relation between simple tensile tests and the behavior of materials observed in practical operations, and it did it by the introduction of an "experience" or safety factor. All efforts of modern research in the field of strength of materials, testing of materials, and analysis of parts has not been able to eliminate completely this safety factor. This could be done only by making calculable in advance all the influences which necessitate its use.

The following of these influences are named here: (1) The difference between the behavior of material in a tensile test and its behavior under a spacial state of strain in a boiler. (2) Deviation of the properties of material as represented by a test bar from the properties in an actual construction which includes defects in manufacture, porosity, non-uniform heat treatment, etc. (3) Lack of precision in the computation of the strains occurring. (4) Differences between the properties of the material which are assumed in the computation of parts and the properties of the material when it undergoes actual strains. (For example, in test computations when dealing with the tensile strength and the elastic limit of materials, while in long-range straining the corrosion effect has to be considered.) (5) Reduction of the resistance of material during operation because of corrosion, changes in the condition of surface, etc. (Symposium in *Zeitschrift des Vereines deutscher Ingenieure*, vol. 78, no. 13, Mar. 31, 1934, pp. 395–401, illustrated. The last article of the symposium, "Numerical Calculation of Long-Range Failures in Operation," has not been abstracted because of lack of space. *etA*)

Short Abstracts of the Month

AERONAUTICS

Internal-Combustion Turbine for Aircraft

THE turbine with which the author deals is of the hot-air type rather than of the explosion type. The principle of the turbine is illustrated in Fig. 1 where a fan-bladed wheel *A*

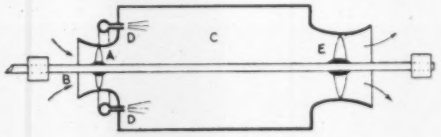

FIG. 1 DIAGRAM OF A CONTINUOUS-FLOW-TYPE HOT-AIR TURBINE

sucks in air through orifice *B* at a temperature T_0 and atmospheric pressure p_1, and delivers it under a certain pressure p' into a combustion chamber *C*, where the air is heated by the nozzles *D* injecting in an atomized form some such fuel as a heavy oil. Because of this, its specific volume increases in a ratio of T' to T_1 where both are expressed in degrees of absolute temperature. In leaving the combustion chamber the hot air expands from a pressure p' to atmospheric pressure and drives the wheel *E* held on the same shaft as the compressor wheel *A* (this wheel also may be on a shaft of its own). It would appear therefore that if the temperature of the air is 15 C (288 C abs) and if the air in the combustion chamber is heated to 300 C equivalent to 573 C abs, the specific volume is nearly double and the expansion from p' to p_a generates an amount of power nearly twice as large as the work of compressing the air from p_a to p' and about one-half of the work of expansion can be converted into useful work. This presupposes, of course, that the system operates without losses in friction, turbulence, etc., but even when the losses are taken into consideration quite a substantial proportion of useful work remains. The author claims that the weight per unit of power generated will be lower in the case of a turbine of this type than in the case of a reciprocating motor. To prove this he considers a tube with the free cross-section of 1 sq dm (15.50 sq in.) delivering into the atmosphere a jet of hot air moving at a velocity of 200 m (656 ft) per sec. The kinetic power of such a jet having a temperature of about 100 F is equivalent to 25 hp. If, however, the jet comes out of the tube not at atmospheric pressure but at a pressure five times as great, as the fluid is then five times as dense, the kinetic power developed will be 125 hp. In turbines a velocity of flow of the working fluid at the rate of 200 m (656 ft) per sec is not at all extraordinary and it may be of interest to note that the kinetic power of a jet increases as the

cube of its velocity. The jet, therefore, will have a power of 1000 hp in the case of a tube of 1 sq dm (15.50 sq in.) discharging into a chamber at a pressure of 5 atm.

The thermodynamics of this problem, including the entropy diagram, have been published by the author in *La Technique Moderne*, for June, 1921, and he points out that certain recent developments such as the Velox boiler furnace are along the lines which he then indicated. The apparatus described does not seem to have been built as yet. (Marcel de Coninck in *L'Aerophile*, vol. 42, no. 1, January, 1934, pp. 14–19, 1 fig., *g*)

AIR MACHINERY

Northey Rotary Air Compressor

THIS machine consists of a main water-cooled chamber *a* (Fig. 2) which is formed by two intersecting cylindrical bores. Within these bores two rotors *b* revolve simultaneously

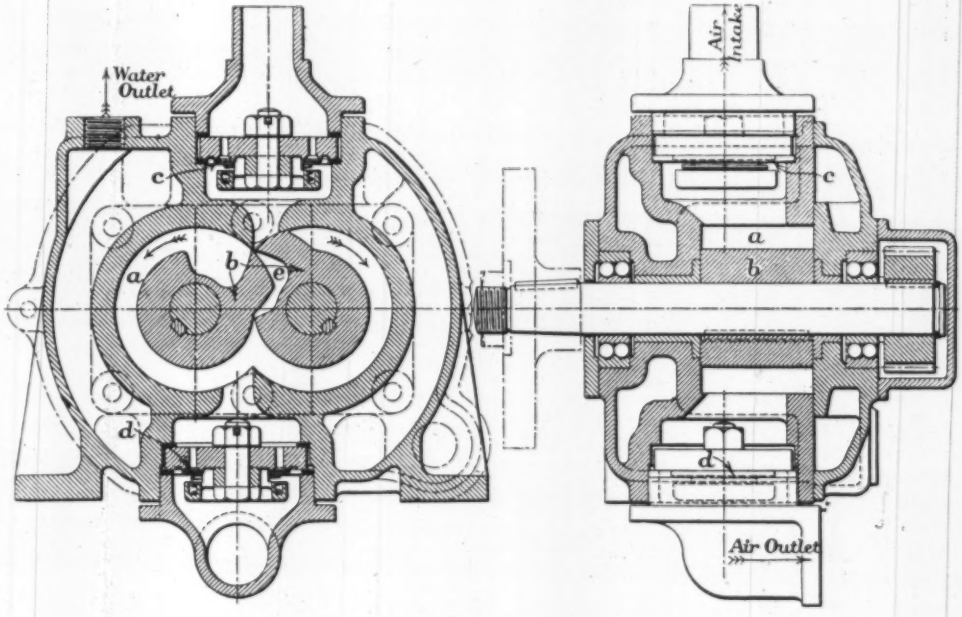

FIG. 2 NORTHEY ROTARY AIR COMPRESSOR

in opposite directions, being carried on two parallel shafts which are concentric with the bores. The shafts are coupled by equal wheels at one end, so that the rotors are maintained in correct angular relationship one to the other and no variation in relative angular velocity can take place between them. They are formed with small clearances so that they do not come into actual contact with one another, or with the walls of the chamber in which they are situated. The rotors revolve in the direction indicated by the arrows in the illustration, and as they move from the position shown in this figure the volume enclosed between the trailing sides of the projections increases and air is drawn through the inlet valve *c*. At the same time, the volume enclosed between the leading sides of the projections is reduced and the contained air is compressed. The end of the compression stroke is reached when the projectors on the rotors reach a position immediately above the delivery valve *d*. The position of the rotors at this stage is similar to that shown in the illustration, except that they are inverted. By the time the rotors reach this position, most of the compressed air will have been expelled through the delivery valve *d*, a small pocket, however, remaining in the space *e* between the rotors.

This clearance volume *e* has no deleterious effect on the efficiency of the compressor. As the rotors move on from the position representing the end of the compression stroke, the two projections move upward, lying between the two rotor shafts. Both valves automatically close and the small volume of compressed air which is contained in the space *e* is released in an upward direction and expands throughout the whole chamber, raising the pressure to a value above that of the atmosphere; that is to say, the chamber is supercharged. When the rotors again reach the position shown in the illustration, practically the whole of this supercharged air will be contained in the space on the leading side of the rotors, that is in the volume which is being compressed. The only part of the charge which must fall to atmospheric pressure before further air can be drawn in is that then contained in the space *e*, and as this is only about 5 per cent of the original clearance air, its effect on the efficiency is negligible. The supercharging effect will naturally be greater the higher the delivery pressure of the machine. It will be clear from this description that the induction and compression stages are produced in a common chamber by means of a continuous and uniform rotary motion, so that no inertia forces have to be considered.

Tests on this machine were carried out by Dr. S. J. Davies, of King's College, London, with machines running either as a compressor or as a vacuum pump. In the course of these tests the compressor ran at speeds of 900 rpm, 1550 rpm, and 2200 rpm, with a range of pressures from 14 to 93 lb per sq in. The base-circle diameter of the rotors of the compressor under test was 2 in., the length $2^{1}/_{16}$ in., and the diameter of the casing $2^{7}/_{8}$ in., so that the radial width of the two annuli was $^{7}/_{16}$ in. and the distance between the rotor center lines 2 in. It is claimed also that a very high vacuum can be obtained with this machine. (*Engineering*, vol. 137, no. 3561, Apr. 13, 1934, pp. 425–426, *de*)

ENGINEERING MATERIALS

Beryllium Copper

SMALL wheels of a beryllium-copper alloy are being used for electrical contacts in a very delicate instrument and have only lately replaced other material; namely, a small wheel of nickel-silver, the rim of which was covered with a reasonable thickness of gold. The beryllium-copper is giving satisfaction because it is hard, has a long life, and sufficient electrical conductivity. (*The Iron Age*, vol. 133, no. 16, Apr. 19, 1934, p. 25, *d*)

FUELS AND FIRING

The "Still" Process for the Production of Liquid Hydrocarbons From Coal

THE "Still" process uses a method of internal suction or extraction. It is characterized by the use of a suitable distribution of the subatmospheric pressure in the interior of the

three zones of the oven charge. The valuable products of distillation, no matter how long the coking time may be, are compelled to pass off through the cool coal, thereby avoiding undue decomposition. By preventing decomposition in this manner increased quantities of benzol and tar are obtained and the quality of the latter is improved. On the other hand, there is a slight decrease in ammonia, coke, and gas formation. The essential features of the plant and processes are given in detail in the original article.

The oven is charged in the regular manner, whereupon spears are thrust down guide pipes into the coal charge and immediately withdrawn. This operation leaves a number of holes in the center of the coal charge. Suction tubes are then inserted into the same guide pipes, the bell on the upper end of the tubes projecting downward into the boxes forming, in conjunction with the guide pipes, a water seal. The lower ends of the suction tubes are inserted into the coal to a depth of about 200 to 300 mm (7.87 to 11.81 in.) and are rendered gas tight by means of the plastic layer.

Two gas zones are formed and a balance between them established. Withdrawal of the gases and vapors from the inner zone is carried out until such time as the plastic layers coalesce in the center line of the oven. When this happens the suction tubes are removed and until the end of the coking process the gases are permitted to pass away in the usual manner through the other zone. Insertion and removal of the suction tubes are operations which are quickly carried out. Two plants are working according to this system and the general figures for their performance are given in the original article.

About 50 to 60 per cent by weight of the tar produced by the Still suction process is of a type suitable for treatment by cracking and a system of cracking has been developed for treating this tar. It is said that the plant has been working for some considerable time with excellent results. Very brief details of this part of the process are given in the original article. (H. Dean, in *Fuel in Science and Practice*, vol. 13, no. 4, April, 1934, pp. 112–115, *d*)

INTERNAL-COMBUSTION ENGINEERING (See also Aeronautics: Internal-Combustion Turbine for Aircraft: Motor-Car Engineering: Engine Design From a Wear and Maintenance Viewpoint)

LUBRICATION

Prospect of Material Changes in the Technology of Lubricant Refining

THE author briefly sketches the demands that lubricating oil of the future may have to satisfy. Oil refiners claim that no better oil is needed and that the present quality is giving excellent service. The author grants all of these arguments and still believes that better oil must be had, because the development of a better oil will be forced by competition. The pressure will not come initially from the consumer or even from the automotive manufacturer, but will be forced by the oil companies' sales departments in the scramble to obtain new selling arguments to meet competition.

In all save possibly one essential step, today's lubricants are made as they were made 40 years ago. In a generation, no radically new ideas have invaded refining practise, unless we except centrifuge dewaxing, itself only a new way of accomplishing an old result. A new plant for carrying out the oldest method of wax removal—by cold settling—was built less than four years ago. The new methods of refining developed these last two or three years have not been rushed upon the refiner. In fact, in principle, none of them is new or unknown.

The author mentions particularly the vacuum distillation of lubrication fractions, the principle of which is old but the details of which may not be. Three major types of processes are speedily changing the entire picture in refining for the immediate future. These processes are solvent extraction and dewaxing, hydrogenation, and synthesis. In solvent refining alone nine units are in operation and seven more under construction on contract, with contracts imminent for several others. There are possible troubles ahead in the way of application of some solvent refining methods. Among other things, the author quotes a prominent refiner who said that those who had bought processes and units are praying that this equipment will prove to be flexible enough to handle any new developments which may come along in the near future. Another potential difficulty which looms large is the possibility of patent tangles.

A factor in the use of extraction solvents is the fact that the extract, or discarded portion of the fraction, is unchanged and may be used for a variety of purposes. From the nature of the material, it may be expected that cracked gasoline derived therefrom would show some advantage in knock rating over gasoline cracked from more paraffinic stock. Edeleanu, in one or two earlier patents, claims the use of such extract from kerosene and gasoline as an anti-knock blending material. It is recognized that the more naphthenic the charging stock, the higher the knock rating of the cracked distillate, other things being equal.

The recent tendency toward the use of lighter motor lubricants brings new problems to the refiner. In addition to this the author claims that more knowledge of lubrication and better test methods are needed. Reluctantly we are forced to admit, says the author, that so far neither the oil-refining nor the automotive industry knows a great deal about the actual mechanism of lubrication. This does not belittle the scientific data which have been collected during the last half-century, but rather recognizes the extent of the knowledge that is yet to be obtained regarding the "how" of lubricating a bearing.

The investigations of Herschel, King, and others have shown that a property of oils aside from mere viscosity must be responsible for much of the lubricant's behavior in service. Mougey, Maag, and others, in studying extreme-pressure lubricants, have shown that, under certain conditions, viscosity has little to do with effective lubrication. That property which has not been identified is usually referred to as "oiliness," the property by which an oil maintains a film under service conditions. The author cites, in this connection, the paper by O. R. King before the Institution of Petroleum Technologists, London, abstracted in MECHANICAL ENGINEERING, Vol. 56, No. 5, May, 1934, p. 292. (Arch L. Foster, in *National Petroleum News*, vol. 26, no. 7, Feb. 14, 1934, pp. 23–28, including bibliography, *cA*)

MACHINE PARTS

The Hilliard Overrunning Clutch

DESIGNED for constant engagement and release, this clutch has a series of rollers running between an outer ring and a series of flat surfaces on an inner hub. The rollers are held in a bronze cage retained in position by a small bracket on the hub. The drive may be applied either through the outer sleeve or the inner hub, and the direction of rotation can be readily reversed in the field. There are two flat surfaces on the hub for each roller, providing new wearing surfaces by moving the hub to a new position within the roller cage. The unit is said to be

sufficiently flexible to serve either as a clutch or as a coupling to connect two shafts. Available in twelve sizes for capacities from 1/4 to 100 hp, at 100 rpm. Also furnished with a trip

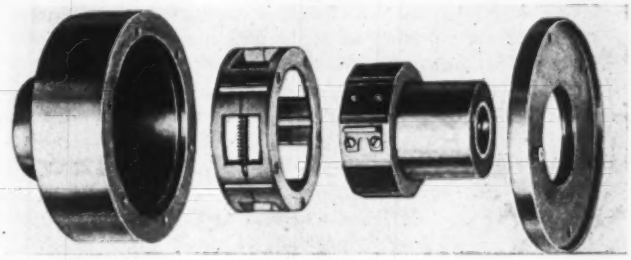

FIG. 3 HILLIARD OVERRUNNING CLUTCH

arrangement for use as an automatic clutch or intermittent drive. (*Product Engineering*, vol. 5, no. 5, May, 1934, p. 194, 1 fig., *d*)

MACHINE-SHOP PRACTISE

Machine Feeds and Cutting Efficiency

THE conclusions presented in this article were prepared in the belief that it was possible to calculate the effect of the feed used on over-all costs. The matter of speed is affected by a number of variables and where a single cutting speed is given it cannot take all the variables into consideration. Consequently it is probably lower than could be used much of the time as it must be suitable for heavy cuts which are not taken more than a portion of the time. It is attempted here to show the way in which variations in the feed in metal-cutting set-ups affect the total cost of the piece produced.

Study of available data on the cutting of metals has indicated that the pioneer work of Frederick W. Taylor is still more complete and reliable than that available from any other source. For this reason the data contained in "On the Art of Cutting Metals" were used as a basis for a new set of tables which represent as closely as possible the cutting speeds which should be attainable with properly hardened present-day tool steels and tools of the size and shape adopted. As regards tools it was thought desirable for the purpose of these tests to adopt a compromise form of tool nose which may be readily formed by grinding and gives satisfactory cutting performance. This tool form, illustrated in the original article, is quite extensively used in England, Germany, and Sweden. It is only 3 1/2 per cent less efficient in cutting performance than the corresponding Taylor tool and has a slightly reduced tendency to chatter. The author states in detail how he obtained the values for the various items that go into the matter of costs for different feeds.

From the curves showing the total processing cost of a part as a function of the feed it appears that this total cost drops rapidly but that the rate decreases as the feed is increased, particularly after a feed of 0.030 to 0.040 in. per revolution has been reached. While the cost continually becomes lower as the feed is increased, it drops only 15 per cent as the feed is increased from 0.060 to 0.100. In view of the difficulties which are usually encountered when attempting to take heavy feeds when turning steel, it will be necessary to have rather complete data as to the equipment available before deciding just what is the best procedure to follow. Incidentally, the cutting horsepower for a cut 3/8 in. deep and 0.060 in. feed is nearly thirteen, so that it would require a machine equipped with at least a 15-hp driving motor to carry this cut, as it requires at least 3 hp to run the lathe idle. The transverse strength of the tool must also be considered.

From these curves it would appear that the machining cost is in some way inversely proportional to the horsepower required to remove the chip. (Robt. C. Deale, Secretary, A.S.M.E. Subcommittee on Metal Cutting Data, in *American Machinist*, vol. 78, no. 8, Apr. 11, 1934, pp. 269–272, 7 figs., *ep*)

MARINE ENGINEERING

Aircraft Carriers

THE author gives a brief history of the development of the aircraft carriers which apparently began soon after the outbreak of the war in 1914. The *Argus*, completed on Sept. 16, 1918, embodied most of the basic features of the present types. One of these was the arresting device to decelerate more rapidly the aircraft after it had landed on the deck, consisting of a number of wires stretched, in this case longitudinally, over the flight deck, and kept in position by light hinged hurdles which were knocked down by the wheel of the machine when passing them. This gear was fitted in carriers until 1925 when it was removed and inclined palisades added at the deck edges to prevent airplanes which had made a bad landing from going over the side. At about the same time the streamlined "island" was introduced to give room for the funnel uptakes and masts. The *Furious* was arranged without the island, that is, as a flush-deck vessel, with horizontal funnel ducts on either side under the flight deck and outside the hangar. The island, however, was reverted to again in the next two carriers *Courageous* and *Glorious*.

The characteristics of the British aircraft carriers as compared with those of other navies are given in a table in the original article. It is interesting to note that the greatest number of aircraft is carried in the American vessels *Saratoga*, *Lexington*, and *Ranger*. The maximum shaft horsepower is installed on the *Saratoga* and *Lexington*, the Japanese *Akagi* being next. The points of design which the author emphasizes deal with the matter of distribution of space and the arresting gear.

In the discussion which followed the presentation of the paper under review, Sir Eustace Tennyson d'Eyncourt pointed out the changes that have been made in aircraft carriers since their construction in order to embody in them the features that experience has shown to be necessary. He believed that similar changes to those made in the British ships had been made in the *Saratoga* and *Lexington* and that the ultimate cost of each was about £10,000,000. The British ships had cost nothing like as much.

The *Hermes* was the only British vessel that had been designed from the start as an aircraft carrier. It suffered from the disadvantage of being too short: a length of at least 700 ft and not 600 ft should be aimed at. A big ship makes a steadier platform than a small one; her pitching is of less amplitude and angle, and this helps in flying on and off.

Wing-Commander T. R. Cave-Browne-Cave said that in aircraft-carrier design high speed was essential and the ability of aircraft to land on them would be increased by the use of a broad flat stern. The provision of trailing screens was much easier with such a stern than with a cruiser stern, as the height from the sea was smaller and the swinging due to rolling less. There was a strong case here for the employment of light internal-combustion engines as propelling machinery on the line suggested by Mr. Ricardo (MECHANICAL ENGINEERING, Vol. 56, No. 3, March, 1934, pp. 175–176). This would obviate having to deal with large volumes of gases and would conform to the employ-

ment of a broad stern and numerous propellers. At the same time it would enable high speeds to be obtained.

J. H. Narbeth described the paper as timely in view of the fact that another aircraft carrier was to be laid down, though the authorities did not allow the design of this ship to be disclosed. He mentioned that while the *Eagle* was built, the demands were made that she should be at least three knots faster than the fastest cruiser in the field, so that she could be sent off to pick up aircraft without delaying the squadron, and that she should have an all-clear deck. The *Eagle* was built with a long thin island on the extreme starboard side. As a result of adaptation, however, there was a general agreement that the *Furious* should, if possible, have an all-clear deck. In the later vessels, however, the island had reappeared, as experiments indicated that such an island could be built without serious interference with the air currents, and that by carrying the funnel up to a considerable height the gases would be discharged into regions through which it would not be necessary for the airmen to pass.

Wing-Commander T. Boothby said that during recent maneuvers in the Mediterranean neither side was able to use its aircraft carriers owing to the bad sea conditions, and that gave rise to doubt whether more could not be done with the airship. The present aircraft carrier had an average tonnage of 19,000, a speed of 27 knots, accommodation for 31 airplanes, and cost £3,300,000. An airship would carry five airplanes and cost £280,000. It also required only five men to look after each airplane as against 27 men on an aircraft carrier.

H. G. Williams pointed out that the conditions with regard to aircraft-carrier design were the same as they were ten years ago. This was only to be expected since the type of aircraft had not altered.

Capt. H. S. Howard said that the American aircraft carrier *Ranger* was at first constructed as a flush-deck ship; experience, however, showed that an island was so desirable a feature of a carrier that she was converted. He apparently stated also that no airplanes were carried on the flight deck of American aircraft carriers, adding that a practise has been inaugurated in the United States of basing carrying capacity on hangar capacity and rating the latter on floor area in combination with the given head room.

In the course of a brief reply the author of the original paper, Sir Arthur W. Johns, said that he was surprised that the extreme ugliness of aircraft carriers had not been mentioned. He said that they had been rightly called the Cinderellas of the fleet, but they were the ugly sisters as well. (Paper before the spring meeting of the Institution of Naval Architects, March 21–23, abstracted through *Engineering*, vol. 137, no. 3559, Mar. 30, 1934, original paper abridged, pp. 387–389 and discussion p. 385, and *Engineer*, vol. 157, no. 4081, Mar. 30, 1934, pp. 323–324, original paper, pp. 327–328, illustrated, *dc*)

METALLURGY

New Steels

A SWEDISH metallurgist is responsible for one rather interesting discovery. He has ascertained that by the addition of the elements silicon and copper to the ordinary nickel-chromium austenitic stainless steels, the corrosion resistance is raised to an extremely high degree, and, at the same time, the resistance to wear is improved. He proposes a slightly larger percentage of carbon than in the ordinary stainless malleable steels. His steel contains about 25 per cent of nickel, together with 5 per cent of silicon, 3 per cent of chrome, and 1.5 per cent of copper.

A steel not without value is one produced for such purposes as connecting rods, gearing, and the like. This contains approximately 1.25 per cent of nickel and 0.60 per cent chromium, though the exact analysis fluctuates a trifle according to the whims of individual manufacturers. The advantages claimed for this steel are superior ductility and toughness, a greater power of withstanding impact shock, a finer grain, and ready response to heat-treatment. Heat-treatment greatly raises the mechanical strength of the material, and at the same time raises the yield point and the impact strength, while the ability of the steel to take deep hardening minimizes variation of strength in relation to thickness of section.

Another new steel designed to stand up fairly well to high temperatures has an approximate analysis of 0.35 per cent carbon, 0.80 per cent chromium, and 2.05 per cent nickel.

Heat-treating a steel in a dilute atmosphere of chlorine is a process which has been invented for the purpose of increasing the corrosion resistance of the surface of the steel.

Some interesting work has been done on steel for boilers. In Germany a boiler has been built up of plates and staybolts made from chrome-molybdenum steel, and another of copper-manganese steel. These have been quite successful, and although the copper-manganese steel costs a little more than the chrome-molybdenum, it is argued that the results warrant further experiment on these lines. (L. Sanderson, *The Steam Engineer*, vol. 3, no. 7, April, 1934, pp. 300–301, *g*)

MILITARY ENGINEERING

Flood Production by Aerial Bombs

THE author considers various types of dams and water reservoirs which can be destroyed by aerial bombardment, thus producing floods. The destruction of an earth levee by aerial bombardment is a simple matter. From the formula of the ordinary demolition mine effect, such as used by military engineers, it is found that 432 kg (952.3 lb) of melinite is sufficient to produce a crater of 12 m (39.3 ft) in diameter. A bomb weighing 800 kg (1763.6 lb) is all that is necessary. This formula, given in detail in the original article, holds good only for horizontal terrain of dimensions that are large as compared with those of the crater. In a levee the effect will obviously be still greater. On the other hand, it should be remembered that while the military engineer can place his charge in the axis of the dam, the aviator has to rely on a hit at an angle.

The next question is how much of a bombardment will be necessary. The objective, that is, the dam in this case, is not a point, but a line, and it does not matter where the explosion occurs. With its danger zone of 12 m (39.3 ft) per bomb, twelve bombs should be sufficient for a strike from the average altitude. An ordinary bombardment squadron ought to be able to carry that much. In view of the importance of the result and the comparatively low cost of the operation, two or three squadrons operating at the proper ceiling may be very usefully employed.

The author proceeds next to the consideration of seasonal reservoirs and destruction of the dams holding them. The resistance to destruction differs greatly in accordance with the method of construction of the dam. The great gravity dams represent an imposing mass. All such types are capable of very considerable resistance to destruction by bombs, be it aerial or artillery, when hit from above, but cannot resist the excess pressures produced by an explosion of a heavily charged bomb in the water near the dam.

General Moisson's formula, widely used for calculating the effect of underwater explosions, makes it possible to calculate

the excess pressure which an explosion in the water near the wall will produce. A 500 kg (1102 lb) bomb carrying a charge of 300 kg (661.3 lb) of melinite produces an overpressure of 138 kg per sq cm (1962.3 lb per sq in.) or 17.3 times the surface pressure at the foot of some of the dams. As in the majority of cases the factor of safety of a dam against excess pressures is only ten, there is without question enough excess pressure to produce destruction of the dam.

In gravity-type dams, however, one should not be too optimistic as to the effects of an explosion, and it is not enough to compare merely the factor of safety and the relative value of the overpressure. The factor of safety of 2 of the gravity-type dams is much less than that of arch dams. The case of a dam subjected to a pressure 22.6 times greater than the surface pressure over a surface of several hundred square meters (several thousand square feet) is very disturbing. However, the effect of this overpressure may be limited to cracks, a partial dislocation, and the production of leaks, which would endanger the existence of the job in the long run, without, however, producing immediate destruction.

Fortunately for the offensive, however, it is not necessary to attack the dam by an explosion at its foot. An explosion at one-third of the depth below the surface of the water will be certain to rupture the upper part of the dam. Moreover, it must be remembered that a 500-kg bomb is not the maximum that can be used as bombs weighing as much as two tons (4408 lb) are available.

In regions rich in hydraulic resources the offensive may be managed in such a way as to multiply the effect. Thus, if the dams in the secondary valleys are destroyed so that the floods take place at the same time, the effect is intensified. The precision required for a synchronous destruction of dams is entirely within the range of possible military operation. The velocity of the flow of waters thus released is not very high.

For this kind of destruction all that is necessary is a 500-kg (1102-lb) bomb equipped with a double fuse for hydrostatic operation with proper control and delayed percussion.

The calculations quoted presuppose that the bomb will explode at least 20 m (65.6 ft) away from the dam. This condition probably does not have to be satisfied but may be assumed here. This does not mean that the surface of the water must be struck 20 m from the dam if the airplane releases the bomb in flying in the general direction of the dam.

If it is admitted that the trajectory in the water is a prolongation of the trajectory in the air and that the bomb has been dropped by an airplane flying at 240 km (149 miles) per hr, the horizontal component of the velocity of the bomb is 67 m (219.7 ft) per sec. Its vertical component is 221 m (624.9 ft) per sec, providing it was dropped from a height of 2500 m (8202 ft), or 171 m (560.8 ft) per sec, if it was dropped from a height of 1500 m (4720.2 ft). Actually, however, the precision of fire is not as great as that, but it can be done from a height of 2500 m (8202 ft) and certainly from one of 2000 m (6561.6 ft). (Ch. Engr. Rougeron, in *Revue de Forces Aériennes*, no. 52, November, 1933, pp. 1203–1208, *p*)

MOTOR-CAR ENGINEERING

German Trends

THE German post-office and express service are rapidly being motorized, thousands of trucks being bought for this purpose. There is also a great demand for trucks for road construction, and numerous six-wheelers, many of them of moderate load capacity, from $1^1/_2$ to 4 tons, are available. Rigid axles are used in the majority of these six-wheelers, the two rear axles carrying the frame through the intermediary of longitudinal leaf springs whose centers are trunnion-mounted on tubular cross members. The drive generally is by two sets of worm gears in tandem with a short intermediate shaft between the two driving axles.

The gasoline engine is losing ground to the Diesel, and, for example, at the stand of the M.A.N. at the recent automobile show in Berlin there was not a single vehicle with the former type of engine. This company incidentally has abandoned its former policy of building six-cylinder engines only and has brought out a new four-cylinder engine of 60 hp and of the air-chamber type. The Maybach Company produced some small units, one being a three-cylinder engine developing 60 hp at 200 rpm, and the other a four-cylinder of 80 hp, both with direct mechanical injection. Krupp has introduced an air-cooled, horizontally opposed, four-cylinder, two-stroke engine of 70 bhp. Approximately 5000 trucks and buses with Diesel

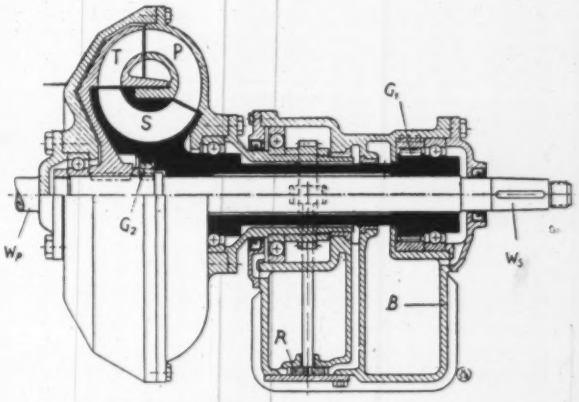

FIG. 4 SECTIONAL VIEW OF A FLUID AUTOMATIC TORQUE CONVERTER

engines were sold in Germany last year and between 11,000 and 12,000 vehicles so equipped are now in operation there.

A considerable business is being done in converting gasoline trucks to the Diesel type, Junkers alone last year sold 300 Diesel engines as replacements for gasoline engines. German railways are coming to use small Diesel engines for switching locomotives. More than 750 of these locomotives equipped with 60-hp Diesel engines are already in service, and some 2000 more will be required. It is also planned to replace many steam trains by rail cars running on a more frequent schedule. For this purpose several new engines of from 150 to 800 hp have been brought out, the latter being the rating of a six-cylinder M.A.N. which delivers this power at 1200 rpm.

The Klein, Schanzlin & Becker Co., of Frankenthal, Palatinate, has developed a fluid torque converter, the manufacture of which has been undertaken by Henschel & Son. The device is a combination of the Foettinger clutch and torque converter and changes automatically from clutch to converter action. Referring to the sectional view shown in Fig. 4, *P* is the impeller, *T* the turbine runner, and *S* the guide blades. When the torque on runner *T* is greater than the engine torque, the blade wheel *S* tends to turn backward. To prevent such backward motion it is mounted on a hollow shaft provided with a free-wheeling unit *G*, at its end. When *S* begins to turn backward the free-wheeling unit locks and *S* then takes up the difference between impeller torque and runner torque. As soon as a balance of torques is established, wheel *S* tends to move forward with the impeller and runner, and it is prevented from overrunning these by a second free-wheeling device G_2 at the forward end, which is then locked. Wheel *S* then acts also

as a runner and adds its torque to that of runner *T*. The entire unit is supported by the housing *B* which contains extra fluid. A pump *R* in this housing constantly supplies fluid to the runner casing in which it is kept under a slight static pressure. This torque converter already has passed through its tests, and a large unit for rail-car use has been ordered by the South African Railways.

Streamlining of long-distance buses has met with favor in Germany. Of particular interest is a Buessing chassis with a V-8 155-hp engine. It is what might be called a $1^1/_2$-decker. The floor is very low and carries two rows of seats facing diagonally forward. The space between these seats is available for luggage purposes. Pillars alongside this space carry a sort of platform, about as high as the seat backs, which forms the gangway for the upper seats, also arranged diagonally but facing inward. There is an air space between the backs of the seats and the body sides so that the upper passengers can look down upon those on the lower deck. The roof is provided with large windows. The bus, of relatively short wheelbase and little more height than a normal single-decker, accommodates 45 passengers and is claimed to be capable of a speed of 75 mph.

Air cooling of engines is gaining ground in Germany year by year. Some years ago the Phaenomen Co., of Zittau, introduced a light truck with a four-cylinder in-line air-cooled engine, which proved a great success. The German post office purchased many hundreds of them for delivery and collection work in country districts. Numerous small "carriers" with air-cooled engines have since been placed on the market. (Edwin P. A. Heinze in *Automotive Industries*, vol. 70, no. 15, Apr. 14, 1934, pp. 450–453, illustrated, *d*)

Engine Design From a Wear and Maintenance Viewpoint

THE author believes that the greatest wear on the greatest number of cars is caused by operating engines at too low a temperature with too light a load factor. While many engines are ruined by overspeeding, he believes that this is a small percentage of the total that require early overhauling due to low-temperature operation and too light a load factor. Cars that show excessive wear are all too frequently the ones that operate almost entirely in city service with a mileage of 40 to 50 miles per day. In the winter months their oil temperature drops to 50 to 70 F at which the dilution, encouraged by a low load factor and the resultant poor compression and incomplete combustion, is not thrown off but remains in the oil and causes trouble.

The recent researches of the Dutch Shell Co. and English investigators point to the corrosive action of the by-products of combustion as the most important factor of cylinder wear. As the temperature of combustion falls rapidly on the power stroke, the layer of gas next to the cylinder wall chills below the dewpoint of the products of combustion and it condenses on the walls. These residual products of combustion contain gases of a highly corrosive nature. Under some conditions with certain fuels, thin films of rust form with incredible speed. Wear may occur at six or eight times the rate observed under more favorable conditions. What can the designer do here to eliminate or minimize the wear due to unfavorable operating conditions?

It would appear that the following items should be given consideration as a means to obtain the desired improvements:

(1) Manifolding that will operate over protracted periods without loading and even give higher mixture temperatures when idling during the frequent stops in city traffic.

(2) Temperature control on the cooling system to assure minimum jacket temperatures of 170 F.

(3) Starting aids that will eliminate the abnormal conditions encountered in the morning start in zero weather.

Many designers calmly shut their eyes to this feature, trusting to a battery A-1 condition, high-test gasoline, a low-viscosity winter oil, and a skilled operator, to secure a start. Unfortunately, this combination is quite uncommon, which accounts for the intense activity of the tow-car between 7:00 and 10:00 a.m. on sub-zero days.

(4) Means for keeping the oil hot in zero weather; either heat applied to the oil by the cooling water or possibly subpans for extreme conditions.

(5) Materials specified for cylinders, pistons, rings, crankshaft, and piston pins that will be more resistant to corrosion than merely average materials.

The wear of valves, guides, and seats is a responsibility that is mainly up to the designer. Location of the valves with relation to one another and the cylinder barrel is possibly one of the designer's most important decisions. Yet at almost every automobile show there can be found models in which valves are spaced so close to the cylinders that deformation of valves and bores adjacent to the ports is bound to show up quickly. Lubrication of exhaust-valve stems and guides is often left to chance, resulting in excessive wear. Engines stopped suddenly under full load at top speed most show oil well up toward the tops of the guides if wear is to be avoided. Exhaust valves are very frequently far too large, often being the same size as the intake valves. They can probably be made 60 per cent of the diameter of the intake valves with no loss in power and a vast improvement in exhaust-valve temperature.

There are two fuel-consumption figures with which the designer should be familiar on both his own and competitive engines. One is the relation of idling consumption to fuel consumption at the load of best economy. The second is the relation of the leanest mixture on which the engine will operate smoothly without missing in any cylinder at wide-open throttle and the setting required for maximum power. These two figures tell in a broad way how well the designer has done his job of manifolding, combustion-chamber design, and spark-plug location.

The designer often overlooks the importance of arranging oil grooves or vents so that such dirt as may enter bearings can be flushed out of the bearing by the oil itself. Oil grooves are often inadequate, and with inferior oil, small grooves and small holes feeding important bearings become closed. Designers think in terms of nice clean oil; not in terms of the diluted, carbon-and-dirt-laden mixture that furnishes the lubrication of the average engine. The designer here must decide what type and size of oil cleaner is sufficient for his engine. No matter what size he chooses, it will be too small.

Still another cause of cylinder wear, while recognized by engine builders for many years, has only recently been receiving active consideration from designers. This is wear caused by explosion pressure getting back of the top ring and forcing the ring out against the barrel. Wear due to this cause is particularly severe in engines which are subject to frequent and rapid applications of load, as in city bus-service where heavily loaded buses must be accelerated many times per hour. Explosion pressures applied to pistons moving at high speeds do not seem nearly so disastrous as the same pressures applied suddenly to pistons moving at low idling speeds. If you wish to show up wear rapidly in cylinders in a test, do not set the engine up and run it constantly at a high speed. Put it on a cycle that changes abruptly from a low idling speed to full load at moderate speed 20 or 30 times per hour, and the wear will be sufficient. (J. B. Fischer, Ch. Engr., Waukesha Motor Co., in *S.A.E. Journal*, vol. 34, no. 4, April, 1934, pp. 132–135, *gA*)

POWER-PLANT ENGINEERING (See also Aeronautics: Internal-Combustion Turbine for Aircraft)

Combination Fuel-Fired and Electric Boiler

IN THE cheese factory at Neuenkirch (Canton of Lucerne) a new type of steam boiler has been put into service which is interesting in that it may be fired with either oil or coal, or used as an electric boiler. The boiler consists of two parts, the fuel-fired boiler and the electric boiler connected to it. When the fuel-fired boiler is in service, the electric boiler may be shut off so that the water contained in it is not heated. On the other hand, when the electric boiler is in service, the water contained in both boilers is heated by natural circulation.

The combined boiler has the advantage of giving very clean service and requiring a minimum of attendance when heated with electric current. If the electric supply is insufficient or is interrupted, the coal or oil firing can be started up at once. In addition, the boiler system can be combined with electric steam accumulators, so that cheap current, available particularly during the night, can be utilized. The load taken by the electric boiler is 150 kw, corresponding to about 450 lb of steam per hr, and this load may be regulated as desired between 30 and 150 kw. The coal-fired boiler is of the vertical type with transverse tubes; it has a heating surface of 55 sq ft and works at a pressure of 70 lb per sq in. (*Sulzer Technical Review*, no. 1, 1934, p. 27, 1 fig., *d*)

Wind Motors

THE Soviet Union has installed an experimental unit for wind-electric generation with the idea of building, if possible, large wind-power plants, 5000 kw and more, as a source of base-load power to be supplemented by hydroelectric plants. This experimental unit was tested in the Balaklawa region of the Crimea, supposed to have winds suitable for such utilization.

The unit consists of a high-speed wind motor rated at 180 hp and an asynchronous 100-kw generator. The generator unit and automatic switching are mounted in the machine room, placed on a structure 25 m (82 ft) high. Electrical energy at 220 volts, three-phase, is transmitted from the generator by a sliding connection and cables to the power house, where it is stepped up to the 6.3-kv network voltage. The unit supplies energy in parallel with a steam-turbine plant of much larger capacity to feed the regional network.

The windwheel, 30 m (98.42 ft) in diameter, has self-regulating variable-pitch blades mechanically regulated by the centrifugal force produced by a change in wind and speed, so that the speed tends to remain constant. The mechanism of this control consists of weights inside the blades resisted by springs, and any change in speed of rotation is transmitted by a system of links and levers to change the position of the ailerons, which, in turn, causes the effective pitch of the blades to change and thus tends to keep the speed and voltage constant. This whole mechanism is in the rotating nose of the unit. Starting and stopping of the wind motor proceeds in the same manner by turning the stabilizing ailerons. The total weight of the structure and unit is 50 tons; the heaviest part, the rotating nose, weighs 8 tons without the blades. The generator is driven by gears meshing with the rotating nose.

The shaft of the windwheel and generator is inclined at 12 deg to the horizontal plane, and the generator is provided with a pivot bearing in addition to regular bearings to carry the horizontal component of the weight of the rotor. The generator is rated at 220 volts, 600 (synchronous) rpm, power factor 84 per cent; low voltage was adopted for the protection of the experimenters. The rotor is phase-wound and, to prevent the generator from being overloaded, has a motor-controlled rheostat slip-regulator in its circuit, which is actuated by a current transformer in the output circuit.

When the unit arrives at synchronous speed the generator is automatically cut in, and after the wind dies down the generator is automatically cut out until the wind rises again. All automatic equipment and relays are operated on alternating current, 220 volts, thus eliminating the cost of direct-current equipment and battery; this auxiliary power is supplied by a house transformer. Phase synchronization is not necessary before the unit is cut in, but only a speed roughly that of synchronism is needed, so a centrifugal mechanism is used to open or close the circuit of a solenoid-operated circuit breaker connecting the generator to the output low-voltage side of the transformer.

The motor has three regulated blades of wood and metal covered with sheet steel, each blade 11 m (36 ft) long, 2 m (6.56 ft) wide at the inside and 1 m (3.28 ft) at the tip. Maximum blade thickness is 0.68 m (26.77 in.). The blades are free to turn radially and to vary the pitch of the blades to insure constant speed as the wind changes. It is said that the total yearly output of the unit is 200,000 kwhr; no information as to costs is given. (*Electrical World*, vol. 103, no. 15, Apr. 14, 1934, pp. 540–541, illustrated, *d*)

The Sulzer Single-Tube Steam Generator

THE article abstracted describes the construction and regulation of this well-known type and trials carried out under the direction of Prof. A. Stodola, of Zurich.

The main feature of the tests here referred to is the regulation of the boiler. The feedwater delivery, and, if necessary, the intensity of firing, are controlled by hydraulically operated regulators which are under the influence of a temperature indicator located at the outlet end of the single tube constituting the steam generator. This regulation is divided into two parts, a "pre-regulation" which is effected by an intermediate thermostat controlling the first part of the tube from the feedwater inlet up to the injection point, and a "fine regulation" by a thermostat at the outlet which influences the temperature from the injection point up to the outlet of the tube.

Tests carried out on a single-tube boiler designed for raising a maximum of 7.5 tons of steam per hour at a pressure of 100 atm gage and a temperature of 400 C superheat, have shown that when suddenly changing the rate of steam raising by about 50 per cent of the maximum, the normal regulating apparatus being in operation, the fluctuation in temperature at the outlet from the boiler amounted to 24 C when reducing the rate of steam raising, and 17 C when increasing it.

Under the same circumstances, with the precision regulation, fitted with two thermostats, fluctuations of only 10 to 15 C were obtained when the rate of steam raising was increased, and $8\frac{1}{2}$ to 19 C when it was being reduced.

Further it was also proved that by suddenly opening the steam outlet valve wide, reductions in pressures from 98 to 62 atm could be caused without the least sign of a "boiling over." In spite of the absence of a collecting drum, and the extremely small quantity of water contained in the boiler, it would be possible, for example, in a marine plant, to allow sudden increases of load of 50 per cent and more to occur, since the pressure would be raised again to the normal within a few minutes, if the boiler is equipped so as to be fired with oil. (A. Stodola, *Sulzer Technical Review*, no. 1, 1934, pp. 1–19, 24 figs., *d*)

RAILROAD ENGINEERING

Wear-Resisting Rails

INCREASED resistance of rail steel to wear has hitherto been sought in the following three main directions:

(1) Rail of ordinary carbon analysis or of alloy steels in which the head, web, and foot are of the same metallic structure.
(2) Compound rails in which the head is of extremely hard steel while the web and foot are of mild-steel quality.
(3) Heat-treated rails.

The manufacture of compound rails requires extreme care both in teeming the metal into the molds and in the rolling, while close attention is also required in cooling in order that the very hard alloy steel in the rail head may not fissure and that strains set up in the rail owing to the presence in it of two steels of widely differing physical properties may not cause failure in service.

Experience shows that freedom from brittleness may be reasonably assured in the case of heat-treated rails when the steel is, in itself, soft and tough. The effect which was formerly produced in soft rails by the work-hardening of the running surfaces can now be produced in advance by heat treatment, by utilizing the valuable characteristic of carbon steel by virtue of which its microstructure and physical properties, particularly its hardness, can be greatly modified by suitable heat treatment. Three processes in particular for the heat treatment of rails have now been perfected on a commercial scale: the British Sandberg process, the French Neuves-Maisons process, and the German process of the Eisenwerkgesellschaft Maximilianshütte, of Rosenberg, Bavaria. The Sandberg and Neuves-Maisons processes effect a relatively smooth and gentle cooling, which produces a sorbitic structure in the head of the rail; whereas the Maximilianshütte process depends on a sharp and rapid quenching of the rail head in a stream of flowing water with a view to producing a martensitic structure—the hardest attainable structure—on the running surface of the rails.

As previously mentioned, the Maximilianshütte process uses the quickest possible cooling of the rail head in order to obtain a maximum degree of hardness by the formation of a martensitic structure, and the rapid quenching in water reduces the temperature of the head of the rail from 820 C to below 200 C in about 1 min. The rail is then removed from the water, and its hardened upper surface is normalized to some extent by heat flowing from the interior of the head, as shown by the rise in the temperature curve, so that there remains a modified martensitic structure of extreme hardness, yet tough enough, it is claimed, to withstand the conditions of railway service. It is probable that as a result of the modification mentioned, this structure approximates more closely to troostite—the next lower structural range—than to the full martensitic condition. The actual process of hardening is described in the original article. The equipment described provides only for the hardening of one rail at a time, but a device for the simultaneous hardening of three rails is briefly referred to.

In an editorial in the same issue of *The Railway Engineer* from which this abstract has been taken, it is told why rails rolled in the earliest days of steel making have stood up so remarkably well. Though the steel was made to analyses which would not have been countenanced at the present time, the low rolling temperatures of the period, and the increased amount of work done on the steel in reducing it to section, imparted to it a combined hardness and toughness which rendered it far in advance of the traffic conditions of its time. Thus these rails lay in the track for years, benefiting all the time from the effect of work-hardening, as the rolling loads passed over them, while at the same time suffering little or no wear. By the time the severity of the running conditions—load, speed, and frequency of trains—had caught up and overtaken the original wearing capacity of these rails, their running surfaces had developed an extremely tough structure, which, as compared with the original structure, had probably gained up to 50 per cent superior resistance to abrasion. Had these early rails been rolled with the cross-sectional area of present-day sections, they might have remained indefinitely in use; and in any event, in subsidiary lines such rails, still in excellent condition, are carrying traffic after forty and in some cases fifty years' continuous service. (Cecil J. Allen in *The Railway Engineer*, vol. 55, no. 3, March, 1934, pp. 91–94, 5 figs., and editorial, pp. 67–68, *d*)

SPECIAL MACHINERY

Ring-Type Cold Strip Mill

THIS mill, built by the Treadwell Engineering Co., of Easton, Pa., is said to be the first successful mill of the kind introduced in the United States. It was originally developed in 1926 in Sweden, and while applicable to all metals, is said to be of particular value in rolling stainless and high-carbon steels down to thin gages, including so-called ribbon strip.

The mill is essentially of three-high type with a baby middle roll. Rolling takes place between the bottom and middle rolls. Instead of transmitting the rolling pressure in conventional fashion from rotating rolls through bearings, chocks, and adjusting screw to the mill housing, which ultimately provides for the necessary reaction to the separating force between the rolls, the rolling pressure is directly transmitted to, and absorbed as internal stress in, two large rings contacting the necks of the large rolls. These rings, of hardened alloy steel, are suspended on the necks of the top and bottom rolls so that they are free to rotate with the rolls. The system of rings and rolls may be likened to a roller bearing where the middle roll corresponds to the shaft, the top and bottom rolls to the rollers, and the rings to the outer race.

The rolls are kept in their correct position by light roller bearings, mounted on the extension of the roll necks, with the chocks located in the windows of a light cast-iron frame. The bearings of the bottom roll are fixed in the frame, those of the middle roll are mounted in a yoke-shaped member, suspended from an adjustable spring device at the top of the frame, while the top roll bearings are only held in a horizontal direction.

By tightening the springs, the middle roll, the top roll, and the ring are lifted upward, until the inner periphery of the rings contacts the bottom roll necks. After contact has been established, a final adjustment of the spring tension causes a certain amount of pressure to prevail in all contact surfaces. This initial stress has the effect of making the whole assembly act like a friction clutch. The bottom roll, being connected to the drive, sets the rings as well as the top and middle rolls in rotation by virtue of the frictional force produced by the initial spring pressure.

Regardless of the magnitude of the rolling pressure, this initial spring pressure will always allow the driving moment of the rolls and rings to be greater than the braking moment, occurring when a strip is inserted in the mill. It is therefore impossible for an uncoupled roll to stall. If the strip refuses to be gripped, it will slide against the rolls, but these will continue to rotate. The design thus provides for positive drive of all rolls with only one roll coupled, rendering the pinion stand superfluous. (*Steel*, vol. 94, no. 21, May 21, 1934, pp. 48 and 50, 3 figs., *d*)

Recent Developments in CONDENSER TUBES

AN INVESTIGATION of test methods now in use for determining the relative resistance of condenser-tube alloys to corrosion is being made in an effort to prepare a set of standards for corrosion testing of condenser-tube materials. As a preliminary step in the investigation it was thought advisable first to review briefly corrosion test methods in order to obtain some information regarding their value.

The principal methods of corrosion testing that have been used in condenser-tube research are simple total immersion, alternate immersion, salt spray, accelerated electrolytic, impingement attack, experimental condenser, and service.

METHODS OF TESTING

The first three methods serve satisfactorily to indicate the relative corrosion resistance of alloys in a medium in a quiescent state or possibly in slow motion, but they have not been found to be sufficiently indicative of relative corrosion resistance when the corroding medium is in a state of rapid turbulent flow, especially in the presence of entrained air such as exists in condenser tubes in service.

Since suitable protective film formation is the basis of condenser-tube life, the electrolytic method apparently is not applicable to predicting suitability of alloys for condenser tubes.

Corrosion testing by impingement attack, designed to study the combined effect of corrosion and concurrent erosion of the protective film, consists in directing a stream of the corroding solution against a flat surface of the metal under test. The temperature and degree of aeration of the solution and velocity of the impinging stream are controlled. This method of testing appears to give a better indication of the manner in which alloys will resist corrosion in condenser-tube service than do the usual types previously discussed.

Small experimental condensers, operating under more or less controlled conditions, have been used as a further step toward obtaining more representative conditions of service in the laboratory. The conditions of operation of these experimental condensers vary widely as the result of the wide variation of operating conditions in the service it is desired to simulate and the large number of operating variables in a condenser, such as temperature and character of cooling water, rate of flow, and degree of aeration. It would be desirable if a standard type of experimental condenser for corrosion testing of condenser tubes could be developed.

Finally, service is the ultimate test for determining suitability of a material for use under any given operating condition but service conditions do not permit, even within an individual condenser, suitable control for standardization of conditions.

The results of the investigation indicate that testing by impingement attack and the experimental condenser operating under controlled conditions offer the most promise for predicting relative service characteristics of condenser-tube alloys.

CHLORINATION

The continued increase in the use of chlorine for minimizing the formation of slime in condenser tubes has increased the interest among power engineers in the value of this method of cleaning. The practical experience, therefore, of engineers who have used chlorine for the prevention of fouling in condensers is of interest, particularly with respect to the extent of corrosion over a period of a number of years.

Reports have been received on 30 installations in which chlorination has been in use on more than 80 condensers for from 1 to 5 years. The reports cover both fresh-water and salt-water installations ranging in size from a single condenser using approximately 5000 gpm of circulating water to an installation having eight condensers with a circulating-water capacity of approximately 750,000 gpm. Residual chlorine concentrations at the condenser inlet vary from 0.03 to 2 ppm.

In a majority of the installations reported on, the chlorine was introduced into the circulating water by conventional chlorinating apparatus of the solution feed type, either with or without limestone towers. Feeding was generally intermittent except in those installations having spray ponds as a source of circulating water, in which case continuous feed was the usual practise. Several cases were reported in which dry chlorine gas was introduced directly into the wet condenser immediately after the unit was shut down and the condenser drained. Results obtained by this method were satisfactory except in the case of one company which reported evidence of slight dezincification of the tubes after using this method for only a few months. Corrosion was probably due to the high concentration of chlorine in contact with the wet tubes.

The effect of chlorine on slime formation has been known for a number of years and a discussion of this phase of the subject is not particularly pertinent to this report. It is interesting, however, to observe that the results obtained from the use of chlorine as a condenser cleaning agent have apparently been very satisfactory. The improvement in condenser cleanliness as evidenced by increased vacuum was reported to be from 0.1 to 0.33 in., with an average gain of approximately 0.2 in.

It was generally reported that there was apparently no evidence of tube corrosion which could be charged to chlorine. In only one case was it reported that there was a definite increase in loss of tube material due to chlorination and in this case the losses as determined from a number of tests varied from approximately 1 to 7.5 per cent additional tube loss per year, so that it was difficult to say how serious a factor it was.

It would appear, therefore, based on experience, in some cases over a 5-year period, that chlorination, as a method of condenser cleaning where slime is the fouling medium, has a material benefit. Where the concentration of chlorine is carefully controlled, the danger of corrosion due to chlorination is rather remote.

END CORROSION-EROSION

Various recommendations have been made from time to time regarding methods of reducing corrosion-erosion due to turbu-

Abstract of Progress Report No. 7 contributed by the A.S.M.E. Special Research Committee on Condenser Tubes and presented at the Annual Meeting, December 4 to 8, 1933, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. The personnel of the committee is as follows: Albert E. White, chairman; Bert Houghton, vice-chairman; Paul A. Bancel; R. N. Barber; William H. Bassett; Donald K. Crampton; Harvey M. Cushing; Harold F. Eddy; Vincent M. Frost; C. S. Gillette; Walter L. Green, Jr.; Charles F. Harwood; C. F. Hirshfeld; Howard W. Leitch; Ernest F. Miller; William B. Price; Horace A. Staples; William Reuben Webster; Robert Worthington.

lence at the tube inlet. A report has recently been received on another method of reducing end-corrosion.

In a condenser installation on the Atlantic seaboard, an observation of tube failure revealed that the maximum wear was usually within the first 6 in. of the inlet end, while the remainder of the tube was in good condition. After 15,300 service hours, all of the plugs were removed from the leaking tubes and the inlet ends of the tubes of both passes were painted for a distance of from 6 to 9 in. with an erosion-resisting cement. When the condenser was ready for service the tubes were tested for leaks and it was found that 772 of the 1317 leaking tubes had been reclaimed for use as a result of painting. The condenser remained in service 5600 hr before it was again painted. As before, all of the plugs were again removed and all of the tubes painted again. In this way the life curve has been kept fairly horizontal at about 7.5 per cent plugged tubes of the total installed. Similar results were obtained on another condenser in this plant.

It would appear that painting the inlet ends of the tubes not only protects them against wear but also has the added advantage of making it possible to reclaim leaking tubes.

ELECTROLYTIC PROTECTION

Reports have been received on the results obtained in three installations of electrolytic protection systems for the prevention of corrosion in condensers. In an installation which was put into operation on a 28,000-sq ft condenser in November, 1926, it is reported that little confidence is placed in the electrolytic protection system.

Another installation of an 18,000-sq ft condenser using salt water uncontaminated by sewage or chemicals was equipped with electrolytic protection in April, 1931, in an effort to reduce pitting and corrosion-erosion of the tubes. It is estimated that tube deterioration has been reduced approximately 75 per cent due to the electrolytic protection system.

In a third case Admiralty tubes in a 20,000-sq ft condenser, using salt and brackish circulating water containing some sewage and chemical pollution deteriorated in two years from corrosion and general dezincification. It is reported that this action was practically eliminated by the installation in April, 1932, of an electrolytic protection system.

From these reports it would appear that in some cases electrolytic protection is of real benefit in reducing corrosion while with others it has no practical value. It would appear that there might be some basic difference in the installation or operation of electrolytic protective systems which would be responsible for the widely differing opinions.

It was felt that current distribution might well be responsible for the varying efficiency of this method of protection and that a series of tests made on a small experimental condenser might bring out information which would be of benefit. The experimental condenser used for this work consisted of eight $^3/_4$-in. brass tubes, 5 ft long, with water boxes at each end. The system consisted of two iron plates as electrodes in each water box placed approximately 3 in. from the tube sheets. The first runs were made circulating sea water at approximately 4 fps.

In view of the fact that current distribution was a feature to be investigated, the first runs were accelerated by reversing the polarity of the protective current. This, of course, corroded the tube rather than the iron plates but the position and degree of corrosion on the tubes indicated the extent to which the current was distributed throughout the condenser tubes.

In run No. 1, in which the set-up was similar to the conventional electrolytic protective system, the slight corrosion of the tubes was at the very ends, indicating that while current was undoubtedly present on the tube sheets and tube ends it did not extend into the tubes. The tubes were, therefore, insulated from the tube sheets in run No. 2 and the mid-points of the tubes were grounded instead of the condenser shell. An inspection of the tubes after run No. 2 showed that the current flow as evidenced by corrosion was extending slightly further into the tubes. In run No. 3 the cathodes were extended through the axes of the tubes by stretching a copper wire through the tubes. A rather uniform corrosion throughout the entire length of the tubes was the result.

Realizing that the experimental runs with reversed current would only indicate comparative current distribution and that it was not entirely comparable with actual installations, the tests were duplicated in a second series of runs using the normal current polarity and accelerating the corrosive action by circulating 2.5-per cent commercial hydrochloric acid through the tubes at approximately 4 fps. Due to the rapid action of the acid on the iron, one brass plate in each water box was used as the anode for the acid runs. A closed recirculating system was used, the acid strength being maintained at 2.5 per cent.

The current distribution as evidenced by the extent of tube protection obtained in the second series of runs, Nos. 4 to 6, confirmed the results obtained in runs Nos. 1 to 3. The extent of tube protection afforded in runs Nos. 4 to 6 was similar in extent to the corrosion experienced in the first three runs, practically the same tube areas being affected by the current in both series of runs.

It is appreciated that accelerated tests are only indicative of what may be expected in operating practise. The circulating water used (dilute hydrochloric acid) made a very severe test. The protective currents used were, however, much higher than those usually found in practise.

The indications from these tests are that the degree of protection varies widely—practically from 0 to 100 per cent—depending on the density of the current flowing into the metal surface to be protected. Protection from corrosion was complete only in the case where current distribution was ideal.

COMPARISON OF ALLOYS

Where the life of the more common tube alloys has been relatively short, an increasing interest has been shown in alloys that are more resistant to corrosion and erosion. Reports have been received which indicate that the results obtained with the newer alloys are satisfactory in some installations while in other cases they are not.

In an installation on the eastern seaboard where Admiralty-metal tubes gave comparatively short service, replacements for the past $2^1/_2$ years have been made with aluminum-brass tubes. It has been reported that there have been no failures of aluminum-brass tubes to date and that there is apparently no evidence of either corrosion or erosion of these tubes. However, an installation of aluminum-brass tubes in the condenser of a tug operating in New York harbor failed after only 12 months.

In a salt-water installation in which some tubes of Admiralty metal, aluminum-brass, and stainless steel have been installed for observation, it is reported that after seven months' operation stainless-steel tubes appear to be superior for this installation. The Admiralty-metal tubes showed the most active corrosion.

Reports have been received which indicate that for some installations copper-nickel-zinc tubes have proved very satisfactory. It is reported, however, that an installation of these tubes in the condenser of a tug operating in New York harbor was found to be very unsuitable. These tubes were also found to be unsuitable for use in the condenser of a ship engaged in North Atlantic service.

A.S.M.E. Boiler Code

Interpretations

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of The American Society of Mechanical Engineers for approval, after which it is issued to the inquirer and published in MECHANICAL ENGINEERING.

Below are given records of the interpretations of this Committee in Cases Nos. 755 (Reopened), and 767 to 771, inclusive, as formulated at the meeting of March 23, 1934, and of Cases Nos. 758, 772, and 773 as formulated at the meeting of April 27, 1934. They have been approved by the Council. In accordance with established practise, names of inquirers have been omitted.

CASE No. 755 (Reopened)

(Interpretation of Pars. H-64 and H-117)

Inquiry: Will it be permissible to connect an automatic low-water fuel cut-off and/or water-feeding control device to the $^1/_2$-in. water-glass connection of a low-pressure heating boiler, provided there is no external use therein of steam or water from the boiler?

Reply: Such a fuel or feedwater control device may be attached direct to a boiler or to the tapped openings provided for attaching a water glass direct to a boiler, provided that such connections from the boiler are non-ferrous tees or Y's not less than $^1/_2$-in. pipe size inserted between the boiler and the water glass so that the water glass is attached directly and as close as possible to the boiler; the straight-way tapping of the Y or tee to take the water-glass fitting, the side outlet to take the fuel cut-off or water-feeding device. The ends of all nipples where used shall be reamed to full-size diameter.

CASE No. 767

(Interpretation of Par. P-288)

Inquiry: Par. P-288*a* calls for every attached superheater to have one or more safety valves near the outlet. Is it permissible to have a safety valve at either end of an outlet superheater header to which steam is delivered from the boiler drum through a plurality of tubes spaced at practically equal intervals throughout the length of the header to provide a uniform flow of steam through the tubes?

Reply: It is the opinion of the Committee that if the superheater outlet header as described has a full free steam passage from end to end, a safety valve or valves located anywhere in the length of the inlet header may be considered as fulfilling the requirements of Par. P-288*a*.

CASE No. 768

(In the hands of the Committee)

CASE No. 769

(Special Rule)

Inquiry: Will boilers and pressure vessels of riveted construction meet the Code provisions if fabricated of an alloy steel plate material conforming to Specifications for CMS (Chrome-Manganese-Silicon) Alloy Steel Boiler Plate which will be designated in the Code as Specifications S-28, having the following chemical and physical properties:

Carbon, maximum, per cent	0.25
Chromium, per cent	0.30–0.60
Manganese, per cent	1.05–1.40
Silicon, per cent	0.60–0.90
Phosphorus, max., per cent	0.035
Sulphur, max., per cent	0.04
Tensile strength, lb per sq in	85,000–100,000
Yield point, lb per sq in	0.55 tens. str.
Elongation in 8 in., min., per cent	1,600,000 tens. str.

The rivet material to be used for fabrication may be of the same analysis except that the carbon shall not exceed 0.17 per cent which corresponds to a tensile range of 75,000 to 90,000 lb per sq in.

Reply: It is the opinion of the Committee that alloy steel plate and rivets having the chemical and physical properties specified and otherwise conforming to Specifications S-28, may be used in the riveted construction of boilers and pressure vessels under Code requirements.

CASE No. 770

(Interpretation of Fig. MA-2)

Inquiry: In Fig. MA-2 of the Code, the definition of W is given as "approximately $1^1/_2t$," whereas in Fig. MA-3 it is given as "approximately $1^1/_2$ in." Should these definitions not be the same and made to read "approximately $1^1/_2$ in." to conform with the corresponding definitions in the Power Boiler and Unfired Pressure Vessel Codes?

Reply: The expression "approximately $1^1/_2t$" which appears in the definition of W in Fig. MA-2 is a typographical error and should read "approximately $1^1/_2$ in." This illustration will be corrected accordingly.

CASE No. 771

(Special Rule)

Inquiry: What are the pressure and temperature limits for bronze safety valves under the requirements of the Code?

Reply: Although the Code contains no direct reference to or specifications for bronze, it is the opinion of the Committee that if non-ferrous fittings or valves are made of a bronze corresponding to A.S.T.M. Specifications B 10-18, or B 61-28, they may be used for temperatures not in excess of 450 F and for any pressure within the temperature limitation.

CASE No. 758

(Annulled)

CASE No. 772

(Special Rule)

Inquiry: Is it permissible under the requirements of the Code, to construct heaters, evaporators, and heat exchangers for power-plant service with seamless drawn Admiralty metal tubes which comply with A.S.T.M. Specifications B 44-24 for Seamless Admiralty Condenser Tubes and Ferrule Stock to form the heat-transfer surface?

Reply: Although the Code contains no direct reference to, or specifications for, Admiralty metal tubing, it is the opinion of the Committee that tubes complying with A.S.T.M. Specifications B 44-33 may be used with safe results in unfired pressure vessels at temperatures not in excess of 406 F. When vessels are so constructed, it is the opinion of the Committee that the official Code symbol stamp for the particular type of vessel may be applied.

CASE No. 773

(*Interpretation of Pars. H-64 and H-117*)

Inquiry: Will it be permissible to attach to a low-pressure steam-heating boiler a low-water fuel cut-off and/or water feeder which embodies a float and float bowl and is so constructed as to permit the flow of steam or water therethrough?

Reply: A low-water fuel cut-off and/or water feeder cannot be attached to a low-pressure steam-heating boiler unless it is so constructed that the water inlet valve cannot feed the water into the boiler through the float chamber and that the device is sufficiently isolated or insulated from the heat to insure a temperature at which scale will not be deposited therein.

BOOK REVIEWS AND LIBRARY NOTES

Hydro- and Aeromechanics

FUNDAMENTALS OF HYDRO- AND AEROMECHANICS AND APPLIED HYDRO- AND AEROMECHANICS. Two volumes based on lectures by Professor L. Prandtl of Göttingen. By O. G. Tietjens. McGraw-Hill, New York, 1934. Cloth, 5¾ × 9 in.; 1st vol., 270 pp., 186 figs., $4; 2nd vol., 275 pp., 311 figs., including 226 figs. in text and 68 on supp. plates, $4.

REVIEWED BY J. C. HUNSAKER[1]

THE Engineering Societies Monographs Committee, acting for the four founder engineering societies in sponsoring these two books, makes available to our engineering profession this comprehensive and authoritative treatment of modern fluid mechanics.

Professor Prandtl has, in the last thirty years, developed the sterile classical hydrodynamics of the nineteenth century into a tool for the engineer. In the hands of his followers, his work has found extensive application in aeronautics, naval architecture, hydraulics, ventilation, and wherever fluids are to be moved. Recognition of the value of his work has been international. In England he was made an honorary fellow of the Royal Aeronautical Society and in this country, honorary fellow of the Institute of the Aeronautical Sciences.

Prandtl's lectures have been for years the Mecca for students of fluid mechanics. Now they are available for study and reference in the English language. These two volumes are, in clarity and elegance of presentation and illustration, a credit to the teacher and a credit to his pupil, Dr. Tietjens. Each is complete in itself, but the two cover the field of fluid mechanics in a complementary manner and are naturally to be considered as companion volumes.

"Fundamentals" treats the mathematical-physical theory of fluid mechanics, including statics of liquids and gases, with discussions of the stability of air masses in meteorology, theory of balloons, surface-tension phenomena, and kinematics and dynamics of non-viscous fluids. The development is rigorous and resembles classical hydrodynamics as treated by Lamb but is distinguished by continuous attention to the physical concepts implied by mathematical statement.

The "Applied" book is primarily devoted to technical applications of the theory of fluid mechanics to real fluids and to the coordination and interpretation of the facts of experience. The treatment is made essentially non-mathematical, except for the chapter on airfoil theory. Engineers will here find experimental material in abundance in generalized form for ready application to problems of flow. This treatment yields a rational condensation of a heretofore bewildering mass of data by extensive application of the laws of similitude, and is a brilliant example of practical simplification derived from a theoretical background.

The volume on "Applications" summarizes the work of Professor Prandtl and his school and as such gives a complete picture of modern fluid mechanics. It brings theoretical considerations and physical interpretation together in such a way that an engineer can readily appreciate this picture and use the conceptions derived from it.

[1] Professor in Charge, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Mem. A.S.M.E.

The New Capitalism

THE NEW CAPITALISM. By James D. Mooney. The Macmillan Company, New York, 1934. Cloth, 6 × 9 in., 274 pp., $3.50.

REVIEWED BY T. M. MCNIECE[2]

PROBABLY at no prior time in our history have as many people been so generally interested in the march of economic events and so hopelessly confused by the course they are taking. That this confusion is not confined to the average man is well indicated by serious disagreement among the specialists.

Underlying all economic behavior is the fundamental law of supply and demand. Production, distribution, real wages, purchasing capacity, and many more of the debatable elements of economic life are manifestations of the inevitable application of this law. Transcending all other influences in this intricate problem is the aggregate importance of family needs. In fact, the word "economics" in its original Greek sense meant "the science of housekeeping." It is the summation of home requirements that provides the objective of the large organized economic activities of America. The same laws govern the economics of home and industry.

This conception is offered in "The New Capitalism" by James D. Mooney and it pictures the everchanging relationships of dynamic forces in a novel manner. In his introduction the author says: "Whatever freshness may be found in this approach really consists in projecting into the discussion of economic problems the discipline of the methods of science; and in using physical analogies as a help in exploring and explaining economic laws." Therein lies a presentation of interest to engineers.

Just as hydraulic analogies have been so successfully used in clarifying to the layman the characteristic behavior of elec-

[2] Scarsdale, N. Y.

trical phenomena, so have they been drawn upon to explain the invisible movements of economic forces. The analyses are clearly and simply made with the aid of a series of graphic charts. In these picture analogies, the interrelations of supply, demand, and prices are shown in terms of the actions of fluids and floating bodies. When the basic law of supply and demand is thus taken apart and put together again, the reader can well realize that this fundamental expression of human behavior can never be repealed by legislative action or nullified by executive decree.

With this illustration of the working of supply and demand as a starting point, the reader is led progressively and always with the same clear analogies through many of the intricate problems of the day. Among these are the medium of exchange, gold, currency and credit, the gold standard, the debt structure, prices, price equilibrium, foreign trade, and purchasing capacity. Production, finance, and distribution are all covered in this analytical manner.

One of the evils in the present plight of the country is that the voter, upon whom the decisive power ultimately rests, and who is necessarily busy earning his own livelihood, has little time to sift the truth from the disagreements of the leaders. Thus, without knowledge of fact, he must cast his vote, if he votes at all, merely as an expression of faith in some one whom he trusts to know more of the subject than he does himself. For those of earnest purpose who really wish to acquire some factual knowledge of the foregoing subjects, the method of analysis followed in this book offers a helpful opportunity.

It is quite probable that one already familiar with the principles so uniquely illustrated in this book may cry, "Too elementary." Possibly so, but the thought persists that some of the sweeping differences of opinion among the experts today might be reconciled by recourse to elementary, scientific methods of analysis.

Out of years of world-wide business experience and fortified by this scientific approach to the baffling questions, Mr. Mooney advances a series of conclusions which show little agreement with many of the present governmental policies. Illustrative of this are the following thoughts taken at random from the text:

> The purchasing capacity of the individual depends not only on his annual income, but fully as much on the cost of things he must buy. . . . The only way to create steady employment is by producing goods continually at a cost that will make it possible to price them to "meet the market."
>
> High prices, particularly if they are out of balance, tend to destroy the interchange of goods, and cause a "starved" flow of goods to the consumer. Such prices destroy purchasing power, dry up demand, create unemployment, and lower the standard of living.
>
> In the ideal economic America, free and equal opportunity would be kept open everywhere and such rewards as are actually accumulated by hard, intelligent work would be protected from appropriation by the government or others.
>
> It is fallaciously believed that a 6-hour day or a 5-day week is a desirable method of spreading employment. Short time spreads employment on a poverty level that soon defeats itself and results in graver employment.

In analyzing the utter fallacy of the idea that we may "work less and have more," pictorial charts are used to show the effect of divided work upon income available for the necessities and conveniences of life. Incidentally, the effect of taxes as a first lien upon all incomes is graphically portrayed.

In the concluding chapter of the book, the author offers a constructive code of procedure to the leaders of each of our important activities and skilfully pleads the case. In addition to the graphic analogies, the book is beautifully illustrated with a wealth of artistic photographs symbolic of many phases of our economic life from farm to city skyscraper. Its strict adherence to scientific principles in the analogies and their interpretations should be refreshing to those engineers who have failed thus far to discover a really scientific analysis of economic laws. This book comes out of the hard school of reality in which the principles set forth are known to work.

Books Received in the Library

AERONAUTICS, Handbook of the Collections illustrating Aeronautics—II, Lighter-Than-Air Craft. By M. J. B. Davy. Science Museum: South Kensington, London, 1934. Paper, 6 × 10 in., 111 pp., illus., 2s 6d. The greater part of this handbook is given to a historical and technical survey. An admirable account is given of the history and development of lighter-than-air craft, from the considerations of Roger Bacon to the designs of today. In this account every salient feature of design seems to be noted, and it can be recommended as an accurate concise summary. An annotated Catalog of the collection of the Science Museum is included in the book.

AIR CONDITIONING. Two Volumes. By S. R. Lewis. International Textbook Co., Scranton, Pa., 1933. Leather, 5 × 8 in., diagrams, charts, tables, Vol. 1, $1.65; Vol. 2, $2.00. The subject of air conditioning is presented simply and clearly in this work, which avoids complicated mathematics and emphasizes practical methods. The physics of air conditioning, equipment, refrigeration, heat transmission through buildings, piping and duct design, and temperature control are discussed, and a final section deals with the design of air-conditioning systems.

AMERICAN INVENTORS. By C. J. Hylander. Macmillan Co., New York, 1934. Cloth, 5 × 8 in., 216 pp., illus., diagrams, $3. The stories of nineteen well-known inventors, from Franklin and Fitch to Jenkins and De Forest, are briefly related in this book, which is intended for boys of a mechanical turn of mind. In spite of their brevity, the sketches are pleasant reading. The personal life of the inventors is brought out and their efforts to attain the goal are clearly shown.

ANGEWANDTE HYDROMECHANIK. Vol. 2. By W. Kaufmann. Julius Springer, Berlin, 1934. Leather, 6 × 9 in., 293 pp., illus., diagrams, charts, tables, 18 rm. The first volume of this treatise, published in 1931, discussed the general theoretical investigations of hydromechanical problems. In this volume, which can be used independently, the author treats certain topics that are of special engineering importance. Discharge from vessels, flow over weirs, flow in pipes and open channels, wave motion, ground-water movement, the hydrodynamic theory of the friction of lubricants, the elements of flow in airplane wings, propellers, and turbines, ship resistance, and wind stresses in structures.

ARBEITEN MIT DEM SCHNEIDBRENNER, Application of the Cutting Blowpipe. (Ausgewählte Schweisskonstruktionen, Vol. 6.) Edited by E. Wiss. V.D.I. Verlag, Berlin, 1934. Cloth, 8 × 12 in., 94 pp., illus., diagrams, 9 rm. The present volume of this atlas of welding practice is planned like preceding ones, the method being to illustrate the uses of the welding torch by photographs and drawings covering a wide variety of actual applications. This section is devoted to metal cutting with the oxyacetylene flame, as used in demolishing buildings and machinery, in fabricating structures, machine parts, etc., both alone and in combination with welding. Brief explanations in English and German accompany the photographs. There are also a short historical introduction and illustrations of many cutting torches and machines. Over two hundred photographs are given, suggesting many possible uses of cutting in building and machine construction.

ARCHITECTURE OF THE UNIVERSE. By W. F. G. Swann. Macmillan Co., New York, 1934. Cloth, 6 × 9 in., 428 pp., diagrams, $3.75. Dr. Swann's book is addressed to two classes of readers: The educated layman without special knowledge of physics or mathematics, who is interested in the broad features of the design of the physical world and in the significance of man's endeavor to understand that design; and to those who have some technical knowledge of the subject but have not meditated extensively upon the philosophical principles involved. The volume is a brilliant feat of exposition, informal in style, and marked by vividness of illustration and clarity of statement. It can be warmly recommended to those interested in the philosophy of modern physics.

BOILER FEED AND BOILER WATER SOFTENING. By H. K. Blanning and A. D. Rich. Nickerson & Collins Co., Chicago, 1934. Cloth, 9 × 11 in., 156 pp., charts, tables, $3. The theory and practise of feedwater treatment are presented in this manual, which aims to meet the practical needs of the operator of small boiler plants who do not have chemists

available. Advice is given on water analysis and the interpretation of results, and on various methods of softening water and the prevention of scale and corrosion. The information is clear and definite, and will be useful to boiler operators.

BUILDING AN ENGINEERING CAREER. By C. C. Williams. McGraw-Hill Book Co., New York and London, 1934. Cloth, 6 × 8 in., 247 pp., illus., diagrams, charts, tables, $2. An interesting textbook, based upon the orientation and motivation courses given at the State University of Iowa. The work aims to give the student a preview of the character of the profession and of its relation to social organization, to assist him to adopt efficient methods of study, to indicate the nature of the engineer's mode of thinking, to afford a historical background, and to indicate future possibilities. The volume can be recommended heartily to beginning students and those considering engineering as a career.

CENTRIFUGAL PUMPS, TURBINES, AND PROPELLERS. Basic Theory and Characteristics. By W. Spannhake, translated from the German by J. B. Drisko. Technology Press, Cambridge, Mass., 1934. Cloth, 6 × 10 in., 328 pp., illus., diagrams, charts, tables, $5. This volume is intended to enable the reader to understand the operation and operating characteristics of existing rotary hydraulic machinery by a presentation of the concepts of fluid mechanics, and to instruct him in the design of these machines. A logical method of analyzing pump and turbine operation is developed and a fundamental procedure for design presented.

DESIGN AND USE OF INSTRUMENTS AND ACCURATE MECHANISM. By T. N. Whitehead. Macmillan Co., New York, 1934. Cloth, 6 × 9 in., 283 pp., illus., diagrams, tables, $3.50. While intended primarily for designers and users of instruments, many of the principles involved are applicable to mechanism as a whole, and especially to light machinery, such as carburetors and magnetos, governors, components of textile machinery, and printing presses. The author discusses various types of errors which may be present, and the causes and methods by which they may be remedied.

DEUTSCHES MUSEUM ABHANDLUNGEN UND BERICHTE, No. 6, PART I, GOTTLIEB DAIMLER in der Geschichte des Kraftwagens zur 100. Wiederkehr seines Geburtstages am 17. März, 1934. By C. Matschoss. V.D.I. Verlag, Berlin, 1934. Paper, 6 × 8 in., 28 pp., illus., 90 rm. Dr. Matschoss has made the centenary of the birth of Daimler, March 17, 1934, the occasion for this interesting essay upon the history of the automobile. This is traced from its beginnings, with emphasis upon the part played by Daimler, whose high-speed internal-combustion engines showed the way to the modern car.

DIE DAMPFKESSEL NEBST IHREN ZUBEHÖRTEILEN UND HILFSEINRICHTUNGEN. (Ergänzungsband to second edition of 1924.) By R. Spalckhaver and A. Rüster. Julius Springer, Berlin, 1934. Cloth, 10 × 13 in., 190 pp., illus., diagrams, charts, tables, 32 rm. Primarily, this work is designed to bring up to date the second edition of the authors' well-known treatise, by recording changes in materials and construction permitted by current German specifications, and advances in our knowledge of boiler practise. In addition, however, the book will be valuable independently, as an account of recent developments and novelties in boiler construction and operation, and a description of advanced practise.

EMPIRICAL DESIGN. By L. D. Hayes. Fourth edition. L. D. Hayes, Morgantown, W. Va., 1933. Cloth, 6 × 9 in., 105 pp., illus., diagrams, charts, tables, $1.50. This book, intended for students of machine design, discusses the design of those machine parts which are made and marketed in many standard sizes of the same general form and proportions. The author treats of screw fastenings, keys and taper pins, shafting and shaft fittings and fixtures, transmission members, and pipe and pipe fittings. Tables, formulas, and curves for proportioning these are given, with methods for deriving and using them.

ENGINE INDICATOR, Its Design, Theory, and Special Applications. By K. J. DeJuhasz. Instruments Publishing Co., New York, 1934. Cloth, 5 × 8 in., 243 pp., illus., diagrams, charts, tables, $3.75. An unusually complete and detailed account of the history, theory, and construction of the pressure indicator is presented in this book by an author with practical experience in its development. Indicators for special purposes, for high-speed engines, for indicating high, mean, and maximum pressures are considered, as well as those of normal type. A chapter is devoted to the analytical theory, and another to testing and calibrating.

ENGLISH FOR ENGINEERS. By S. A. Harbarger. Third Edition. McGraw-Hill Book Co., New York and London, 1934. Cloth, 5 × 8 in., 314 pp., $2. The course presented here is well designed to give the engineering student a good, practical command of the language for letter writing and for preparing reports and technical articles. At the same time cultural reading is not overlooked, and sensible suggestions are given to guide the beginner. The text is a sound guide, concise yet comprehensive, and full of suggestions for further study.

ENTSTAUBUNGS- UND LÜFTUNGSFRAGEN IN DER WERKSTATT. By R. Nagel. V.D.I. Verlag, Berlin, 1934. Paper, 6 × 8 in., 21 pp., illus., diagrams, 1.80 rm. This pamphlet presents the fundamentals of factory ventilation and dust removal in a concise, practical way. It is intended to give the factory manager a general view of the subject which will enable him to devise methods for his own needs. The illustrations show actual installations.

EXPLORING THE UPPER ATMOSPHERE. By D. Fisk, with introduction by H. L. Brose. Oxford University Press, New York, 1934. Cloth, 5 × 8 in., 166 pp., charts, tables, $1.75. A popular description, for non-technical readers, of recent developments in stratospheric physics. The author describes the investigations of Professor Piccard and others by means of balloons and discusses sound as an explorer, ultra-violet rays and their discovery of the ozone layer, the paths of radio waves, polar lights, and cosmic rays. Professor Brose contributes an introduction which sums up our knowledge in more scientific fashion.

FORSCHUNGSHEFT 365. WASSERUMLAUF IN DAMPFKESSELN. By E. Schmidt, P. Behringer, and W. Schurig. V.D.I. Verlag, Berlin, 1934. Paper, 8 × 12 in., 23 pp., illus., diagrams, charts, tables, 5 rm. In recent years, the circulation in water-tube boilers has been under investigation at the mechanical laboratory of the Danzig Technical High School. Certain results of these studies are presented here in three papers. The relative speed with which steam bubbles rise in tubes of various diameters, at pressures up to forty atmospheres, has been measured. Other experiments studied the circulation in a model boiler and the motion of mixtures of liquid and gas in tubes, and measured the heat transfer to such mixtures at various speeds and pressures. In another contribution, the motion of liquid-gas mixtures is discussed theoretically.

GYROSCOPIC STABILIZATION OF LAND VEHICLES. By J. F. S. Ross. Edward Arnold & Co., London; Longmans, Green & Co., New York; 1933. Cloth, 6 × 9 in., 172 pp., diagrams, charts, $5.25. This investigation here reported was undertaken to determine whether monorail traction is scientifically sound and definitely practicable, to show why inventors have hitherto been only partially successful, and to place the whole subject on a more scientific footing. Treating the problem as essentially one of dynamics, the book provides a detailed analytical determination of the principles involved and discusses the considerations that must govern practical design. The work of other investigators is described and a critical review of the literature given. The volume is an important addition to the literature on the subject.

JAHRBUCH DER BRENNKRAFTTECHNISCHEN GESELLSCHAFT E.V. Vol. 14, 1933. Wilhelm Knapp, Halle (Saale), 1934. Paper, 8 × 11 in., 53 pp., illus., diagrams, charts, tables, 5.80 rm. The proceedings at the 1933 annual meeting of the Society are presented in this volume. Power economy in German traffic, the economic importance of steam power, electricity and Diesel engines in railway operation, and recent developments and national problems of Diesel engines were discussed.

MACHINE DESIGN. By L. J. Bradford and P. B. Eaton. Third edition. John Wiley & Sons, New York, 1934. Cloth, 6 × 9 in., 289 pp., illus., diagrams, charts, tables, $3. The object of this text is to supply a brief course which can be covered in about twenty-five lessons, and which will emphasize the fundamental facts and processes of machine design. The new edition has been revised to conform with recent developments, especially in lubrication and welded construction.

MACHINES AUTOMATIQUES, MÉCANIQUES ET ÉLECTRIQUES. (Collection Armand Colin.) By P. Maurer. Librairie Armand Colin, Paris, 1934. Paper and cloth, 5 × 7 in., 185 pp., illus., diagrams, 10.50 fr.; bound, 12 fr. A concise study of the automatization of machines. In its first chapters, bases for a classification of automatic machines and of the general theory of mechanisms are discussed, after which the processes for making machines automatic are considered. Succeeding chapters discuss the chief automatic electrical and mechanical devices and the principles that govern them.

MECHANICS OF ENGINEERING. By S. D. Chambers. Macmillan Co., New York, 1934. Cloth, 6 × 9 in., 279 pp., diagrams, $3.50. This work is a class-room text covering only the essentials of statics and kinetics, intended for students of engineering who are acquainted

with the elements of the calculus and college physics. The material is arranged to fit one-hour recitations, highly symbolized mathematical proofs are avoided, and numerous illustrated examples and problems, based on the necessities of general engineering practise, are given.

MITTEILUNGEN AUS DEN FORSCHUNGSANSTALTEN G. H. H. KONZERN. Vol. 3, Part 1, April, 1934. V.D.I. Verlag, Berlin. Paper, 8 × 12 in., pp. 1–28, illus., diagrams, tables, 3.15 rm. This bulletin contains four communications upon recent research work in the laboratories of several German industrial concerns. Dr. Requa presents some considerations upon similitude in turbines. Mr. Becker finds the common cause of breakage of the piston rods of double-acting two-cycle engines to be the cooling water. The generation of water-gas in coke ovens is discussed by Dr. Heckel, and Dr. Rademacher reports investigations upon enameled stove castings and their testing.

MYSTERIES OF THE ATOM. By H. A. Wilson. D. Van Nostrand Co., New York, 1934. Cloth, 6 × 9 in., 146 pp., diagrams, tables, $2.50. After outlining the classical view of matter and electricity which was current in the nineteenth century, Professor Wilson describes how new discoveries have required old ideas to be discarded and replaced by a series of new ones. Chiefly, he concerns himself with those about the microscopic structure of the universe, but large-scale phenomena are considered in chapters on cosmic rays, relativity, and gravitation. By omitting technical details, the story has been told briefly, yet with great clarity. The book is a model of compression.

PETROLEUM PRODUCTION ENGINEERING, Oil Field Development. By L. C. Uren. Second edition. McGraw-Hill Book Co., New York and London, 1934. Cloth, 6 × 9 in., 531 pp., illus., diagrams, charts, tables, $5. This edition has been extended to two volumes, of which the present is devoted to oil-field development. The problems of petroleum exploration are sketched briefly, after which the planning and conducting of field development are discussed in detail, and the methods and equipment used in drilling are described. The work aims to cover petroleum production up to the point where the wells are ready to produce and is intended as a text-book for engineering students and a general reference work for executives and engineers.

PHYSICAL OPTICS. By R. W. Wood. Third edition. Macmillan Co., New York, 1934. Cloth, 6 × 9 in., 846 pp., illus., diagrams, charts, tables, $5.25. A clear, scholarly, and thoroughly modern treatment of its subject is afforded by this volume. As in preceding editions, Professor Wood has avoided the use of intricate mathematics and has endeavored to give physical pictures whenever possible of the processes usually described by equations. Special attention is also given to describing experimental technique. Nearly one-half of the old edition having become obsolete the book has been largely rewritten, and over one-half is entirely new. The book can be recommended heartily.

PRESSEN VON NICHTEISENMETALLEN. By W. Brunnckow. V.D.I. Verlag, Berlin, 1934. Paper, 6 × 8 in., 20 pp., illus., diagrams, tables, 1.60 rm. Hot pressing, cold pressing, and die casting of non-ferrous metals are presented briefly in this pamphlet, which is intended to show the possibilities of press working. The machines and tools are described, as well as the properties of the non-ferrous metals and alloys, as a guide to the selection of process and material.

PROTECTION BY PATENTS OF SCIENTIFIC DISCOVERIES, Report of the Committee on Patents, Copyrights, and Trade Marks. (Occasional Publications of the American Association for the Advancement of Science, No. 1, January, 1934. Supplement to Science, Vol. 79.) By J. Rossman, Chairman, F. G. Cottrell, A. W. Hull, and A. F. Woods. Science Press, New York. Paper, 3 × 10 in., 40 pp., $0.50. This report discusses very fully the problem of the scientific worker whose researches lead to results of commercial value. The first division of the report considers the advantages and disadvantages of protecting by patent inventions of industrial and medical value, and the patent policies of universities, the Federal government, and other institutions. In part two, the Committee discusses the desirability of legal protection for scientific discoveries. Both questions are considered at length by the Committee, composed of Messrs. Joseph Rossman, F. G. Cottrell, A. W. Hull, and A. F. Woods, and their conclusions given.

ROLL PASS DESIGN. Two volumes. By W. Trinks. Second edition. Penton Publishing Co., Cleveland, Ohio, 1934. Cloth, 6 × 9 in., diagrams, charts, tables, Vol. 1, 201 pp., $4.65; Vol. 2, 246 pp., $6.15. Although only a short time has passed since this work first appeared, the edition has been exhausted and a new one made necessary, and the author has seized the opportunity to revise and extend the work. Volume one discusses the principles underlying the design of rolls and roll passes; in volume two the theories advocated are tested and enlarged by studying the rolling of the sections in common use. The book is based on extended study and experience, and is easily the most important work in English on its subject.

ROSTFEUERUNGEN. By W. Marcard. V.D.I. Verlag, Berlin, 1934. Cloth, 6 × 8 in., 138 pp., illus., diagrams, charts, tables, 9.50 rm. Methods of firing coal in boiler furnaces are systematically presented in this work, the first on the subject to appear in Germany for many years. The theory of combustion and the factors for determining the value of a fuel, and the processes of ignition and combustion in boilers are first discussed. The author then treats the important problems of grate and furnace construction, after which the construction and operation of modern grates and stokers are considered. The work gives a concise, practical account of good current practise.

ROYAL TECHNICAL COLLEGE JOURNAL, Vol. 3, Part 2, January, 1934, pp. 205–334. Royal Technical College, Glasgow. Paper, 7 × 10 in., illus., diagrams, charts, tables, 10s. 6d. This Journal is devoted to recording research work by the staff and senior students of the College. Papers of special interest to engineers in this number include: An example of magnetic aging, the determination of the moisture content of wood, two applications of the thin conical-wall theory, parabolic weirs, viscous effects in dynamometer belts, electrode systems for dielectric loss measurements, and some notes on the ignition coil.

DER SCHLEUDERGUSS. By A. Väth. V.D.I. Verlag, Berlin, 1934. Paper, 6 × 8 in., 107 pp., illus., diagrams, charts, tables, 6.90 r.m. After a brief historical introduction, the author discusses the forces entering into centrifugal casting, the problem of liquation, and the influence of centrifuging upon the grain and physical properties of the metal. A chapter is devoted to molds. The second section deals with practical methods and apparatus for casting pipe, and for making castings of iron, steel, and non-ferrous metals. There is a useful bibliography.

TATSÄCHLICHER UND GEMESSENER FEINHEITGRAD GESCHLIFFENER FLÄCHEN. (Institut für Schleif- u. Poliertechnik der Technischen Hochschule Braunschweig, Part 2.) By A. Lindau. Friedr. Vieweg & Sohn Akt.-Gesellschaft, Braunschweig, Germany, 1934. Paper, 7 × 10 in., 42 pp., illus., charts, tables, 2.80 rm. The investigation records the development of a new instrument for determining the roughness of finished surfaces, by which accurate numerical values for the degree of fineness can be obtained, instead of merely comparative ones.

THEORIE UND PRAXIS DER SCHWINGUNGSPRÜFMASCHINEN. By W. Späth. Julius Springer, Berlin, 1934. Paper, 7 × 10 in., 98 pp., illus., diagrams, charts, tables, 12 rm. This book is intended as an exhaustive yet brief account of the various questions connected with the carrying out of vibration tests and the evaluation of the results. The first section is devoted to theoretical considerations of vibration, machines for testing, methods of measuring vibration, and of evaluating resonance curves. The second part discusses practical uses of vibration testing, including measurements of bridges, ships, soils, buildings, vehicles, and machinery.

UMBAU VON WERKZEUGMASCHINEN. Grundsätze wirtschaftlicher Umbauten von gebrauchten Werkzeugmaschinen. Part 1, Spangebende Maschinen, by A. Schroeder. Part 2, Maschinen für spanlose Formung, by W. Meier. V.D.I. Verlag, Berlin, 1934. Paper, 6 × 8 in., 21 pp., illus., diagrams, charts, tables, 2.40 rm. An interesting pamphlet describing methods of rebuilding machine tools in order to increase their output. Practical advice is given upon the directions in which betterment may be sought, with illustrations of a large number of improved lathes, milling machines, and other tools, which show changes actually made.

WAVE MECHANICS: Advanced General Theory. By J. Frenkel. Clarendon Press, Oxford, England; Oxford University Press, New York, 1934. Cloth, 6 × 10 in., 524 pp., tables, $12. This work continues the comprehensive treatise begun with the publication, in 1932, of the author's "Wave mechanics: elementary theory." This volume is devoted to the mathematical development of the ideas underlying the new mechanics, to connecting it with classical mechanics and constituting it a complete self-supporting theory. A third volume, dealing with special problems, is to follow.

ZWEITES REFA-BUCH, Erweiterte Einführung in die Arbeitszeitermittlung. Edited by Reichsausschuss für Arbeitszeitermittlung. Beuth-Verlag, Berlin, 1933. Cloth, 6 × 8 in., 122 pp., diagrams, charts, tables, 4.60 rm. This volume, issued by the National Commission for Time Study, is a concise, practical handbook for the manager and industrial engineer. The fundamental principles of time study, the technique of securing job data, and their analysis and synthesis are discussed.

WHAT'S GOING ON

I. A. WINTER

L. N. MCCLELLAN

P. A. KINZIE

This Month's Authors

THE three authors of this month's papers on the Boulder Dam are engineers with the United States Bureau of Reclamation at Denver, Colo.

Phillip A. Kinzie, who writes on the cylinder gates in the intake towers of the Boulder power plant, is a native of Denver. For fourteen years he was connected with the Leyner Engineering Co., of Littleton, Colo., and at the time he left to enter the Bureau of Reclamation, in 1920, he was chief engineer of that company. At present he is senior engineer in charge of the design of gates and valves in the mechanical-engineering section of the Bureau.

L. N. McClellan, author of the paper on the mechanical features of the Boulder power-plant generators, was graduated from the University of Southern California in 1911 with the degree of B.S. in electrical engineering. That summer he entered the Bureau of Reclamation where he has served ever since with the exception of a period of service in the Corps of Engineers during the War and a year in the operating department of the Southern California Edison Co. Since 1924 he has been chief electrical engineer of the Bureau of Reclamation in responsible charge of design, construction, and operation of all power plants, pumping plants, transmission lines, and substations built and operated in connection with Federal irrigation projects.

Ireal A. Winter, a native of Tennessee, writes on the mechanical features of the Boulder power-plant turbines. He began his engineering work on the drafting board in 1912 at the by-product plant, Fairfield, Ala., of the Tennessee Coal, Iron & Railroad Co. Prior to the War, during which he served in France with the 37th Engineers, he was connected at various times with the Alabama Power Co. and Ford, Bacon, and Davis. Following the War he served with several engineering and construction companies and from 1931 to 1933 was a member of the firm of Gilfillan and Winter, consulting engineers. He accepted his present position with the Bureau of Reclamation in 1933 and has been engaged on engineering designs of the mechanical installations for the Boulder power plant.

President Doty Appointed HOLC Supervisor

COL. PAUL DOTY, president A.S.M.E., has been appointed Regional Supervisor of the Home Owners' Loan Corporation for Alabama, Florida, Georgia, North Carolina, and South Carolina, with headquarters at Atlanta, Ga. The HOLC administers the distribution of some two hundred million dollars to be used in modernizing and reconditioning homes. Twelve regional supervisors, each with a background of architectural experience, have been appointed.

Engineering Societies' Secretaries Visit Washington

ON JUNE 4, at the invitation of the Executive Secretary of the American Engineering Council, the secretaries of the national engineering societies were privileged to spend the day in Washington, visiting the government departments which have the most direct relation to engineers and engineering development. Calvin W. Rice, secretary, and C. E. Davies, executive secretary, represented the A.S.M.E.

DINNER WITH LOCAL ENGINEERS

During the day, the secretaries visited the Department of Labor, the Department of Interior, the NRA, the Russian Embassy, and the Department of Commerce. In addition, John Frey, vice-president of the American Federation of Labor and member of the Labor Advisory Board of the NRA, met with the secretaries at the office of the American Engineering Council. At luncheon were the Hon. Frederick A. Delano, chairman, and Mr. Charles Eliot, II, executive officer, of the National Planning Board. At dinner the presidents and secretaries of the local sections of the respective engineering organizations were present and discussed broadly the questions of national policies affecting the membership, including employment stabilization, professional development, the engineers' code, and the upholding of civil-service regulations in local, state, and Federal departments. Maj. Philip R. Faymonville, a member of A.S.M.E., who sails for Russia on June 20, as military attaché at the United States Embassy at Moscow, U.S.S.R., was a guest at dinner.

LABOR AND PWA PROBLEMS

At the Department of Labor, Dr. Isadore Lubin described in detail the organization of the Bureau of Labor Statistics, of which he is the chief, and his new plans for securing more active data on employment, not only in the wage-earning groups, but in the professional groups. At the Department of Interior, Col. H. M. Waite, Deputy Administrator of Public Works, outlined broadly the present situation with regard to public works, emphasizing particularly the fact that there were on hand applications for loans under the Public Works Act, which might be considered suitable, of approximately two and a half billions of dollars. He expressed the hope that money would be found for the mapping and surveying project, which has been promoted at the request of the state and local engineering groups, as well as the national societies, by the American Engineering Council.

PLANNING BOARD AND NRA

Emphasis at the luncheon with Messrs. Delano and Eliot was placed on the difficulties of, as well as the opportunities for, national planning. Both Mr. Delano and Mr. Eliot stated that the purpose of the present commission was really to provide a plan for a plan. Mr. H. R. Glancy, Division Administrator, Capital Goods and Machinery Codes, and Major George Berry, Division Administrator, Construction Codes, discussed the broad implications of the present NRA policies with regard to these groups. They stressed the idea that the job of compliance with codes can only be made possible through effective trade organization on a voluntary basis and indicated that the trend in code administration lay in putting the burden for compliance on the organized trade groups. They outlined also the process of application through which the new code must go and indicated that the present disposition was to try to relate new applications for new codes to some code authority or code machinery already established.

SECRETARY ROPER EXPRESSES THANKS

The day's visits closed with a conference with the Secretary of Commerce, the Hon.

Daniel C. Roper, who expressed his thanks to the many members of the organizations who had served on advisory groups to the Department of Commerce. He expressed the hope that this type of cooperation would be looked upon as a fundamental in the development of his department and said he would welcome any suggestions as to the way the department could serve engineers and engineering.

Coal-Utilization Course at Illinois

THE College of Engineering of the University of Illinois has announced a short course in "Coal Utilization," to be given July 12, 13, and 14. The purpose of the course is to give technical information in a non-technical way that concerns not only the coal operator and combustion engineer, but salesman, purchasing agent, and retail coal merchant. The course covers the whole field of the utilization of coal and will deal primarily with preparation and combustion and the use of combustion equipment. The morning sessions will be devoted to lectures and the afternoon sessions to discussions by prominent men in various phases of the coal industry, representing coal-production, retailing, sales, purchasing, and combustion-engineering departments of producers, retailers, and consumers.

A.S.M.E. Petroleum Division Meets at Tulsa

THE Petroleum Division and the Mid-Continent Section of The American Society of Mechanical Engineers held a four-day meeting at Tulsa, Okla., May 14 to 17, in connection with the Eighth International Petroleum Exposition. The program of the six technical sessions, which were devoted to refining, production, and transportation, was printed in the May, 1934, issue of MECHANICAL ENGINEERING. The attendance at the technical sessions exceeded two hundred.

William G. Heltzel, of the Stanolind Pipe Line Company, and chairman of the A.S.M.E. Petroleum Division, opened the meeting with an address of welcome and was followed by William B. Way, general manager of the Petroleum Exposition, who expressed his gratification at the cooperation of the A.S.M.E. in holding its meeting in connection with the Exposition. Some of the technical papers were preprinted by the *Oil and Gas Journal* and *Petroleum Engineering*, the publishers providing a sufficient quantity of preprints for distribution at the meeting. The demand for these preprints was so large that no copies are now available, but members of the A.S.M.E. interested in reading the papers are referred to the *Oil and Gas Journal* and *Petroleum Engineering*.

Two inspection trips were provided for the fourth day of the meeting. At the Hominy Pumping Station of the Stanolind Pipe Line Company, visitors were given an opportunity to see the actual transposition of the 6- and 8-in. pipe lines to 12-in. lines and were shown how the cast-iron pipe which has been in use for 15 or 16 years, is salvaged, and held as a supply for future small-line developments in new fields. The Hominy Pumping Station is an example of excellent housekeeping. The second trip included visits to one of the stations of the Texas Pipeline Company where modern equipment of large capacity is very compactly installed; to the Engineering School of the University of Oklahoma, at Norman, where research on flow through pipe is being conducted with the cooperation of the A.S.M.E., members of the faculty, engineering students, and certain C.W.A. workers; and to a new well in the neighborhood of Norman which is being drilled with Diesel engines.

At the Petroleum Exposition the A.S.M.E. again exhibited its automatic pumping station which has been completely rebuilt and brought up to date through the efforts of engineers in the mid-continent area and the cooperation of the number of companies who very generously loaned the necessary equipment. The station has been greatly simplified since the original was built for the 1930 Exposition.

The week's program was completed with a meeting of the A.S.M.E. Mid-Continent Section on Friday evening, when students from the University of Oklahoma, the Oklahoma A. & M. College, the University of Arkansas, and the University of Tulsa presented papers.

During the course of the meeting there was an informal gathering of the various engineering educators from eight colleges. The new policy governing student branches of the A.S.M.E. was discussed and the hope was expressed that the policy would be extended to the colleges represented at the meeting, all of which would be in Group 6.

Langley Field Conference

THE Ninth Annual Aircraft Engineering Conference of the National Advisory Committee for Aeronautics, held at Langley Field, Hampton, Va., May 23, 1934, was as interesting and representative as usual. The group of visitors included such men as Orville Wright and Col. Charles Lindbergh. Prof. Alexander Klemin, New York University, represented The American Society of Mechanical Engineers and MECHANICAL ENGINEERING at the conference, and prepared the following report.

While nothing sensational or highly novel was disclosed at the conference, various demonstrations and brief reports submitted by the committee's scientific staff gave continued evidence of solid fundamental work which will have its influence on every phase of aircraft-engine design and construction.

In the aircraft power plant initial attempts in the forced cooling of radial, finned engines by the use of blowers were noted. It has often been said that air-cooled engines are severely limited in capacity; the blower leads to the possibility of an almost indefinite increase in specific output for such types.

Experiments on safety fuel indicate that a high compression ratio of 9.5 to 1 is feasible with this fuel, so that both output and efficiency are now on a par with those obtained with gasoline, while at the same time the safety of a heavier fuel will be available to aircraft operators.

In aerodynamics the committee is very wisely taking a forward step by building a 20-in.-throat high-speed tunnel in which airfoils will be tested at speeds approaching those of the velocity of sound. Fundamental work on the effects of high speed is of obvious importance in the design of the modern propeller and in the investigation of stratosphere flight. Research in a small high-speed tunnel has already resulted in the development of a high-speed airfoil which decreases the loss due to compressibility at high speed.

The immense full-scale tunnel is busily at work, and for the first time in the history of aeronautics tests at the Reynolds number of full flight can be made under controlled conditions.

The committee has continued its researches on rotary-wing aircraft, and members of the staff are of the opinion that the high-speed possibilities of such aircraft are not negligible. The cyclogiro[1] has been tested with satisfactory results in static lift, although forward performance has fallen somewhat below theoretical anticipation.

All visitors rode on the carriage of the towing basin, whose length is 2200 ft. The speed of this trip was limited to 25 mph. The maximum speed is 56 mph (but is apparently too harrowing to the uninitiated). The tank is busily employed on a systematic investigation of the many factors which enter into float and flying-boat-hull design, such as the effects of a pointed step, the variation of ratio of step depth to hull width, the angle of rise, etc.

A comprehensive, mathematical, and experimental study of airfoil flutter is under way in which there are, unfortunately, 12 parameters to confuse the issue.

This is but a brief outline of the immense work carried on by the committee. It is gratifying to report continued progress and persistent and systematic effort.

Drawings and Drafting Room Practise

THE proposed American Standard for Drawings and Drafting Room Practice is now before the members of the Sectional Committee on Drawings and Drafting Room Practice for vote on approval by letter ballot. This proposed standard is to be considered at the Annual Meeting of the S.P.E.E., joint sponsor with the A.S.M.E. for this project, to be held at Cornell University, June 19–23.

Referendum on Public Works

AT THE request of President Coleman, and as a result of a mail vote by the Executive Committee, two questions were submitted to the American Engineering Council with regard to the policy of the Council in regard to Public Works. The questions and answers are as follows:

[1] See MECHANICAL ENGINEERING, vol. 55 (1933), pp. 376–377 and 421.